This is a digital copy of a book that was preserved for generations on library shelves before it was carefully scanned by Google as part of a project to make the world's books discoverable online.

It has survived long enough for the copyright to expire and the book to enter the public domain. A public domain book is one that was never subject to copyright or whose legal copyright term has expired. Whether a book is in the public domain may vary country to country. Public domain books are our gateways to the past, representing a wealth of history, culture and knowledge that's often difficult to discover.

Marks, notations and other marginalia present in the original volume will appear in this file - a reminder of this book's long journey from the publisher to a library and finally to you.

Usage guidelines

Google is proud to partner with libraries to digitize public domain materials and make them widely accessible. Public domain books belong to the public and we are merely their custodians. Nevertheless, this work is expensive, so in order to keep providing this resource, we have taken steps to prevent abuse by commercial parties, including placing technical restrictions on automated querying.

We also ask that you:

+ *Make non-commercial use of the files* We designed Google Book Search for use by individuals, and we request that you use these files for personal, non-commercial purposes.

+ *Refrain from automated querying* Do not send automated queries of any sort to Google's system: If you are conducting research on machine translation, optical character recognition or other areas where access to a large amount of text is helpful, please contact us. We encourage the use of public domain materials for these purposes and may be able to help.

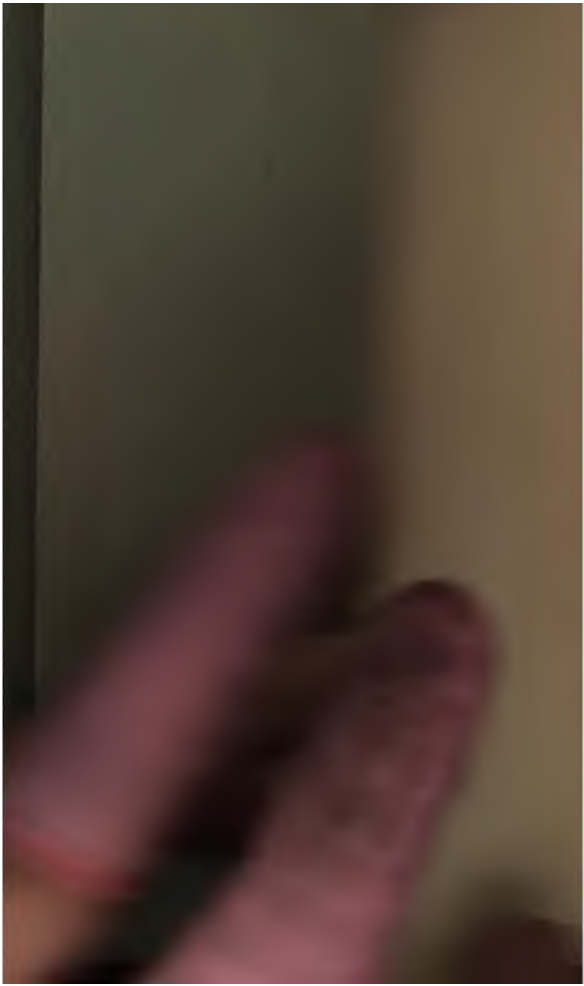
+ *Maintain attribution* The Google "watermark" you see on each file is essential for informing people about this project and helping them find additional materials through Google Book Search. Please do not remove it.

+ *Keep it legal* Whatever your use, remember that you are responsible for ensuring that what you are doing is legal. Do not assume that just because we believe a book is in the public domain for users in the United States, that the work is also in the public domain for users in other countries. Whether a book is still in copyright varies from country to country, and we can't offer guidance on whether any specific use of any specific book is allowed. Please do not assume that a book's appearance in Google Book Search means it can be used in any manner anywhere in the world. Copyright infringement liability can be quite severe.

About Google Book Search

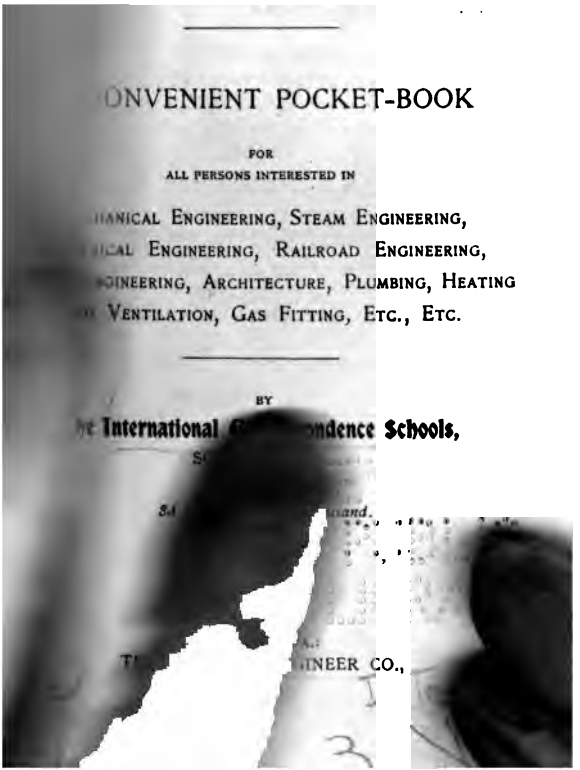
Google's mission is to organize the world's information and to make it universally accessible and useful. Google Book Search helps readers discover the world's books while helping authors and publishers reach new audiences. You can search through the full text of this book on the web at `http://books.google.com/`

NYPL RESEARCH LIBRARIES
3 3433 06272853 4



MECHANICS'

KET MEMORANDA.



NEW YORK
LIBRARY
393495A
LENOX AND
TILDEN FOUNDATIONS
R 1928 L

ENTERED ACCORDING TO THE ACT OF CONGRESS,
IN THE YEAR 1897, BY
THE COLLIERY ENGINEER CO.,
IN THE OFFICE OF THE LIBRARIAN OF CONGRESS,
AT WASHINGTON.

BURR PRINTING HOUSE,
NEW YORK.

PREFACE TO THIRD EDITION.

It is with feelings of great satisfaction that we once more announce a new edition of the "Mechanics' Pocket Memoranda." One of the principal advantages of our pocket-book has been its small size, thus making it a pocket-book in reality instead of in name only. This feature has been retained in the present edition. Notwithstanding that the pocket-book is three times as large as in the preceding edition, it is but a very little thicker, owing to our having eliminated the blank pages at the end. Much new matter has been inserted, and we particularly call attention to the sections relating to Civil Engineering; to Electrical Engineering; to Plumbing, Heating and Ventilation; and to those useful facts for Contractors, Carpenters, etc., which are contained in the section on Architecture.

Respectfully,

The International Correspondence Schools,

SCRANTON, PA.

JUNE 1, 1897.

3 X 2 O

PREFACE TO SECOND EDITION.

The first edition (2,000 copies) of the "Mechanics' Pocket Memoranda" was issued last October, with the hope that it would prove useful to persons connected with Mechanical industries. Its claim to originality was that some of the principles given were stated more clearly and concisely than in more pretentious pocket-books. The popularity which it achieved was unexpected, and has induced us to issue a second edition in which a number of pages containing matter not strictly technical have been eliminated and the space devoted to practical Mechanical information. Considerable new matter relating to Architecture and Electricity has also been added.

We trust the present edition will prove even more satisfactory than the preceding one, and that Engineers, Draftsmen, Machinists, Pattern-makers, Blacksmiths, Foundrymen, Electricians, Architects, Builders, Carpenters, Plumbers, Steam Fitters, etc., will find it convenient and useful.

JUNE 1, 1894.

INDEX.

A.

	PAGE
Air inlet pipes	233
" , Velocity of flow of hot	255
" , Vitiation of	253
Alloys	14
Ampere	148
" turns	170
Annunciator systems	190
Applications of Electric Motors	173
Armature, Bursting of	174

B.

Batteries, Various chemical	175–178
" , Storage	179–180
" " , Regulation of	179
Beams, Bending Moments in	44
" , Cantilever	44
" , Deflection of	44
" fixed at both ends	44
" , Safe loads on	209–210
" , Simple	44
" , Standard connections for	215
Bearing power of soils	196
Bearings for line shafting, Distance apart of	116
Belting, Leather, Arc of contact of	73
" , " , Horsepower transmitted by	72
" , " , Pulleys for	126
" , " , Width of	72
" , Rope, Power transmitted by	130
" , " , Tension in	131
Bell Towers	227
" Wiring	188
Bending moments in beams	44
Birmingham Wire Gauge	164
Blasting in granite	197
Blow, Force of a	69
Blue Prints, To make	93
" " , Paper	93
Boilers, Kitchen	243
" , " , Connections for	245
" , Steam	94
" , Dome of, To develop	89
" , Foaming and Priming of	98
" , Horsepower of	78
" , Inspection and care of	94
" , Prevention of scale in	96
" slope sheet, To develop	90
Bolts for cylinder heads	136
" for steam chests	136
" , Standard proportions of	8

	PAGE
Bricks, Hollow	202
" , Size of	198
Brickwork, Effluorescence on	201
" , Strength of	196
Building Materials	214
" , Heating of	256

C.

Cables, Carrying capacity of	159
" , Testing of (electrical)	188–187
Calendar, Perpetual	316
Candlepower	156
Capacity of cables	159
Cast iron beams, hollow, Safe load on	216
Centrifugal force	51
Chains and ropes	49
Change gears	103
Chimneys	98
" , Formulas for	99
" , Table of sizes of	100
Chord of circle	39
Circle, Area of	16
" , Chord of	39
" , Circumference of	16
" , Segment of	39
Circuit, Derived or shunt	152
Circular pitch, Formula for	71
" " , Table of	72
" rings, Area of	39
" " , Volume of	40
Clearance of piston	135
Clock Towers	227
Coefficient of elasticity	43
" " expansion, Cubic	87
" " " , Linear	87
" " " , Surface	87
Coils for Dynamo	155
Columns, Formulas for strength of	46
" , Wooden, Strength of	206
Commutator, Sparking at	171
Compass Surveying	265–268
Compound gear lathes, Screw-cutting with	107
" pulley, Formula for	68
Compressive strength of materials	43
Conductor, Direction of motion of	152
" , Size of	157
Concrete	198
Conductivity, Electrical	150
Cone, Formulas for	40
Conical frustum, Formulas for	40

PAGE

Connecting rod brasses 143
" " cotters 143
" " straps 143
Contact, Arc of, (in rope gearing) 133
Continuous current 158
Corliss engine crank shaft 142
" " cylinder 134
Corrosion of boilers 96
Corrugated sheet iron 214
Cotters for connecting rods 143
Couplings, Flange 117
" , Flexible 118
" , Proportions of 118
" , Shaft 117
Crank shafts for Corliss engines 142
" " " high speed engine 142
Crossover tracks 314
Cubic expansion, Coefficient of 87
Current, electrical, Rules for direction of 150
" , Strength of 149
" , Induced 157
" , Continuous 158
Curves of saturation 169
Curves, Deflection angles of 275
" , Degree of 275
" , Tangent distance 276
" , To lay out with transit 277
" , " " " without transit 279
" , Elevation of railroad 300
Curving of rails 298
Cut nails 220
Cylinders, Proportions of 135–138
" , Strength of, Formulas for 49
" , Surface of 40
" , Stuffing box for 142
" , Volume of 40
Cylinder heads 135
" " , Bolts for 135
" walls 139
Cylinders for Corliss engines 134

D.

Decimal equivalents of parts of one inch 25
Decimals of a foot, Equivalent in inches of 25
Deflection of beams 44
" " ropes 134
" , Tangent and chord 278
" , " " " , Formulas for 286
Derived circuits 152
Designs of machine details 115

	PAGE
Development of boiler dome	89
" " " slope sheet	93
Diagram, Slide Valve	111
Diametral pitch, Formula for	71
" " , Table of	72
Differential pulley	68
Dome of boiler, To develop	89
Double movable pulley	67
Draft of chimneys, Formulas for	99
Drainage systems, Inspection and Testing of	238
Drilling in granite	197
Drills, Twist, Speed of	102
Duty of pumps	80
Dwelling house walls, Thickness of	194
Dynamo design	167
Dynamo-electric machine	154
" " , Underwriters' rules for	193

E.

Earthwork, Calculation of	295
Eccentric rod	146
" sheave	146
" strap	146
Efficiency, Lamp	160
" , Motor	162–172
Elastic limit, Table of	43
Electro magnet, Polarity of	151
Electric dynamo machines	154
" motors, Application of	173
Electricity	148
Electric gas lighting	181
Electro deposition	181
Electromotive force	148
" " , Formula for	167
Elevation of railroad curves	300
Ellipse, Formulas for	38
Emery wheels, Speed of	103
Engine horsepower, Formula for	76
English and Metric measures, Conversion tables of	29
Equivalent decimal parts of one inch	25
" " " " " foot	25
Exhaust port	136
Expansion, Coefficient of	87
" , Cubic	87
" , Linear	87
" , Surface	87

F.

Factories, Thickness of walls of	195
Factors of Safety, Table of	42

PAGE

Failure of dynamo 171
Falling bodies 50
Feed water, Testing of 97
Field magnet 168
" " , Reversal of 171
Fireproof construction, Cost of 204
Flagging, Transverse strength of 204
Flange coupling 117
" pipes 134
" , Proportions of 118
Flexible coupling 118
Flexure, Ultimate strength under 43
Floors, Loads on 196
" , Concrete work for 198
" , Glass for 213
Flow of water in pipes 84
Fluxes for soldering and welding 15
Foaming of boilers 98
Force, Electromotive, Formula for 167
" , Magnetizing 168
Force of a blow 69
Forces, Resultant of 67
Formulas, How to use 30
Foundations, Concrete work for 198
Frustum of cone, Formulas for 40
" " pyramid, Formulas for 41
Frog (railroad work) 301
" , Angle 302
" , Distance 305
" , Crotch or middle 315
Fusion, Latent heat of 88
" , Temperature of 88

G.

G^2, Values of 45
Galvanized boilers, Sizes of 244
Galvanometer 183
Gas burners 251
Gas fittings 246
Gas, Flow of, Measuring 248
Gas meters, How to read 250
" " , Pressure in 250
Gas, Pressure of 247
" " " , Measuring 248
" pipes, Capacity of 251
" " , Testing 252
Gases, Weight of 13
Gauge sizes of wire, with equivalent sectional areas 163
" , Birmingham wire 164
" , B. & S. wire 164

	PAGE
Gearing, Formulas for	71
Gears, Change for screw-cutting	103
" , To calculate speed of	74
Gibs for connecting rods	143
Girders, Wooden, Strength of	207
Gland	147
Glass for floors	213
" " skylights	213
" " windows	213
Gold, Imitation of	14
Granite, Blasting and drilling in	197
Gravity, Center of, To find	51
Grindstone, Speed of	103
Gyration, Center of, To find	61
" , Radius " , " "	61
" , " " " , " " experimentally	63
" , Square of least radius of	45

H.

Hangers, Shaft	124
" , " , Design of	125
Hardness of woods	212
Heat	87
" , Latent, of fusion	88
Heating of buildings	256
" and ventilation	252
" of dynamos	171
" surface of boilers per horsepower	79
" " , Ratio of to grate area	79
Helix, Formula for	39
" , To construct a	39
High speed engines, Crank shaft for	142
Hollow cast iron beams, Safe load on	216
Horsepower of belts	72
" " boilers	78
" " electrical currents	149
" " engines	76
" " pumps	80
" " rope belting	131
" , Theoretical	75
Hot air, Flow of	255
" water heating	262
" " " , Size of pipes for	263
Hydrokinetics	82
Hydromechanics	80
Hydrostatics	80

I.

I, Values of	45
Imitation of gold	14

PAGE

Incandescent lamp data 160
" wires 191
Inch, Equivalent decimal parts of 25
Inches and parts thereof expressed in decimals of one foot 25-27
Indicated horsepower of engines, Formula for 76
Inertia, moment of, To find 54
" " " " " experimentally 63
" " " " " For various sections 45
Inclined planes, Formula for 68
Incrustation in boilers 96
Interior wiring 155
Inspection of boilers 94
Iron bars, Round and square, Weight of 7
Iron pipes, used as columns, Safe load on 217

J.

Joint Coupling, Universal 118
Journal box, Design of 119-123
" " , Proportions of 119

K.

Keys for shafting, Proportions of 116
Kilowatt 149

L.

Land Surveying 265
Lamps, Efficiency of 160
" , Incandescent, Data 160
" in series (electric light) 155
Lap, Inside and outside 110
Latent heat of fusion 88
" " " vaporization 88
Lathe, Change gears of 103
" , Compound geared 108
" , Simple geared 104
Law, Ohm's 149
Lead of valve 110
Lead, Sheet, Weight of 7
Leclanche cell 190-191
Leakage, Magnetic 170
Leather for grooves of rope sheaves 133
Legal Ohm 148
Length of steam ports 136
" " nails and tacks 218
Levers 66
Leveling, Direct 281
" notes, How to check and keep 283
" , Profiles in 285
" , Grade lines in 285

PAGE

Line shafting ... 116
Linear expansion, Coefficient of ... 87
Lines of force, Leakage of ... 170
" " " , Number of ... 167
Lock nuts ... 115
Loads on floors ... 196

M.

Machine design ... 115
" tools, Cutting speeds for ... 101
" , Motors for ... 175
Magnetic field ... 175
Manilla rope belting ... 130
" " " , Weight of ... 130
Magnetic meridian ... 266
Masonry ... 198
Materials, Strength of ... 41
Maxims in Sanitation ... 234
Mean effective pressure ... 76
Measures and weights ... 5
" " " , Metric ... 28
Mechanical powers ... 66
Mechanics ... 50
Mensuration ... 37
Metals, Weight of ... 9
Metric and English measures, Conversion table for ... 29
Meridian, Magnetic ... 266
Meridian, True ... 266
Mil ... 155
Mil circular ... 155
Moments, Bending ... 44
Moment of Inertia defined ... 54
" " " , To find experimentally ... 63
" " " of various sections ... 45
" " Resistance defined ... 64
" " " of various sections ... 45
Mortar ... 199
Motors, Electric, Application of ... 173
" for machine tools ... 173
" , Efficiency of ... 172
" , Polarity of ... 152
" , Underwriters' rules for ... 193

N.

Nails ... 218–220
" for slating ... 219
" , Weight and length of ... 218
Neutral axis ... 65
Nuts, Proportions of ... 8
Nut-locks ... 115

PAGE

O.

Oblique fixed pulley, Formula for 68
Ohm, Legal 148
Ohm's Law 149
Oil cup 121
Oscillation, Center of, To find 57
" , Radius of, To find experimentally 63
Output of Motors 172

P.

Painters' memoranda 228
Paper, Blue print, To make 93
Parallelogram 38
" of force, Explanation of 67
Passage, Steam 136
Pedestals, Design and proportions of 118-123
Percussion, Center of, To find 59
Perpetual calendar 316
Pipes, Drainage, Fall of 234
" , " , Size of 232
" , " , Weight of and fittings for 229
" , Brass 231
" , Lead 232
" , Wrought iron and steel 230
" for house fixtures, Size of 233
Piles, Foundation 224
Pillars, Wooden, Strength of 206
Pipe flanges 134
Pipes and cylinders 49
" " fittings 229
" , Flow of water in 84
" , Wrought iron, Sizes of 6
Pistons 142
Piston rings 143
Pitch of bolts in cylinder heads 136
" " " " steam chest covers 136
" , Circular, Formula for 71
" , " , Table of 72
" , Diametral, Formula for 71
" , " , Table of 72
Plaster, Cost of 205
Polishing wheels, Speed of 103
Polygon of Force, Explanation of 67
Port, Exhaust 136
" , Steam 136
Power transmitted by leather belting 72
" " " rope " 131
Powers, The mechanical 66
Pressure, Mean effective 76
Priming of boilers 98

PAGE

Prints, Blue, How to make..... 93
Projectiles..... 50
Prony brake..... 172
Properties of Timber..... 211
Proportions of flange coupling..... 118
" of keys..... 116
" of journal boxes..... 119–123
" of belt pulleys..... 127–130
" of rope " 132–133
" of shaft hangers..... 124
Pulleys, Compound, Formula for..... 68
" , Differential..... 68
" , Fixed, Single..... 67
" , Movable, Double..... 67
" , " , Quadruple..... 67
" , " , Simple..... 67
" , Belt..... 126
" , " , Proportions of..... 127–130
" , Rope..... 131
" , " , Proportions of..... 131–133
" , Speed of..... 74
Pumps, Discharge of..... 80
" , Duty of..... 80
" , Horsepower of..... 80
Pyramid, Formulas for..... 41
" , Frustum of, Formulas for..... 41

Q.

Quadruple movable pulley, Formula for..... 67

R.

R, Values of..... 45
Radius of gyration, To find..... 61
" " " , " " experimentally..... 63
" " " , " " for various sections..... 45
" " oscillation, " " experimentally..... 63
Rate of transmission of electricity..... 148
Radii and deflections, Table of..... 286–288
Radiation of heat..... 257
Radiators..... 258
Radiating surface in steam heating..... 258
Reciprocal of a number..... 150
Rectangle, Formula for..... 38
Resistance, Electrical..... 148
" of derived circuit..... 152
" of pure copper wire..... 167
" , Moments of..... 64
" , " " , For various sections..... 45
Resultant of forces..... 67
Reversal of field..... 171

PAGE

Retaining walls 289
" " , Resistance of, To overturning 291
Ribs for cylinder heads 140
" " piston 143
" " steam chest cover 139
Ring, Formula for 39
" , Circular, Formula for 40
Roofs, Angles of 206
" , Coverings for, Weight of 206
" , Shingles for 225
Root of any number, Rule for extracting 34
Rope belting 130
" " , Power transmitted by 131
" " , Pulleys for 133
" " , Tension in 131
" , Manilla, Weight of 130
Ropes, Wire 133
" , " , Pulleys for 133

S.

Sag of rope 134
Safety, Factors of, Table of 42
Safety valves 70
Sanitary Maxims 234
Saturation curves (electrical) 169
Scaffolding with round spars 221–223
Scale in boilers, Prevention of 96
Screw-cutting, Change gears for 103
" , Formulas for 68
Screw-threads, Proportion of 8
Shaft couplings 117
" hangers 124
" " , Design of 125
Shafting, Formulas for 49
" , Line 116
Shearing strengths, Table of 43
Sheaves for rope gearing 133
Sheet, Boiler slope, To develop 90
Sheet lead, Weight of 7
Shingles 225
Shunt circuit 152
Simple geared lathe, Screw-cutting with 104
Single fixed pulley, Formula for 67
Size of bricks 198
" " copper wire for circuits 156–160
Skylights, Glass for 213
Slates 219
Slide Valve 110
" " diagram 111
Slope sheet of boiler, To develop 90

	PAGE
Soils, Bearing power of	196
Soldering, Fluxes for	15
Solders	15
Soil pipes	230
Span of ropes	134
Sparking at commutator	171
Specific gravity, Table of	9
" heat, Table of	87
Speed, Cutting	101
" of emery wheels	103
" " gears, To calculate	74
" " grindstones	103
" " polishing wheels	103
" " pulleys, To calculate	74
" " twist drills	102
Sphere, Formula for	40
Spiral, Length of	41
Standard beam connections	215
" pipe flanges	135
Steam chest	135–140
" " bolt	136
" " covers	140
" " , Ribs for	140
" pipe area	138
" , Velocity of	138
" heating	257
" " , Size of pipes for	259
" " " " boilers for	259
" heating by exhaust steam	261
Steel, Tempering of	15
Stone, Weight of	10
Storage batteries	179–180
Strands in wire rope	133
Strap, Eccentric	146
" end of connecting rod	143
Strength of brickwork	196
" " corrugated iron	214
" " materials (various)	41
" , Transverse of flagging	204
Stroke of engine	135
Stuffing box	142
" " , Proportions of	147
Surcharged walls, Pressure on	293
Surface expansion	87
Surveying with compass	265
" " Transit	268
Switch	300–308
" , Point or split	304
" , Stub	304
" , " , To lay out	308

PAGE

System, The Metric 27
Systems, Annunciator 190

T.

Tacks, Weight and length of 218
Tangent and chord deflections 278
Teeth of wheels 71
Temperature of fusion 88
" " vaporization 88
Tempering steel 15
Tensile strength of materials 43
Tension of rope belting, Formula for 131
Testing of cables (electrical) 183–187
Theoretical horsepower, Formula for 75
Threads, Screw, Cutting 103
" " , Proportions of 8
Timber, Properties of 211
Tools, Machine, Cutting speeds for 101
" , " , Motors for 175
Torque 173
Towers, Bell and clock 227
Tracks, Crossover 314
Track work 298
Transit notes, How to keep 280
Transit surveying 268
Trapezium and trapezoid, Formula for 38
Triangulation 272
" , Inaccessible points 273
" , Measuring inaccessible lines 272
" , Obstacles in line of survey 274
Tunnel sections 294
Turnouts 301–304
" , To lay out 306
Type metals 14

U.

Ultimate strength of materials 43
Units, Electrical 148
Universal joint coupling 118
Underwriters' line wire 165
" , Rules of, For incandescent wire 191

V.

Valves, Safety 70
" , Slide 110
" , " , Diagram of 111
Vaporization, Latent heat of 88
" , Temperature of 88
Velocity of steam 138
" " flow of hot air 255
Vent pipes 230
Ventilation, Forced 254

PAGE

Ventilation and heating ... 252
Vitiation of air ... 253
Volt ... 148

W.

Walls of cylinders ... 139
" " houses and factories, Thickness of ... 194–195
Warehouse walls, Thickness of ... 195
Waste pipes ... 230
Water, Feed, Testing of ... 97
" , Flow of, In pipes ... 84
" , Pipes in buildings, Size of ... 243
Waterproofing of brick walls ... 201
Water supply ... 240
" " , *Hydraulic ram* for ... 241
" " , Pumps for ... 240
Watt, The unit ... 149
Wedge, Formula for ... 68
Weight of copper wire ... 158
" " corrugated iron ... 214
" " manilla rope ... 130
" " nails and tacks ... 218
" " bar iron, round and square ... 7
" " sheet lead ... 7
" " various substances ... 9
Weights and measures ... 5
" " " , Metric system of ... 28
Welding fluxes ... 15
Wheatstone bridge ... 182
Wheel and axle ... 66
Wheels, Emery, Speed of ... 103
" , Polishing, " " ... 103
Wheelwork, Formulas for ... 66
Whitewash ... 201
Width of belts, Formula for ... 72
Window sash weights ... 212
" glass ... 213
Wire, Copper, Sizes for circuit ... 158
" , " , Weight of ... 156–161
" , Incandescent ... 191
" , Steel rope ... 133
" , " " , Strands in ... 133
" , Underwriters' line ... 165
Wiring, Bell ... 188
" Interior ... 155
Wire nails ... 220
" " card ... 218–219
Wood construction, Cost of ... 203
Work, Definition of ... 69
Wrist pin brasses ... 143
Wrought iron pipe, Sizes of ... 6

MECHANICS' POCKET MEMORANDA.

WEIGHTS AND MEASURES.

Avoirdupois or Commercial Weight.

16 drams.....1 ounce.
16 ounces.....1 pound.
14 pounds.....1 stone.
28 " ....1 quarter.
4 quarters.....1 cwt.
2240 pounds.....1 ton.

Square Measure.

144 square ins..1 square foot.
9 " feet..1 " yard.
30¼ " yards.1 " rod.
40 " rods..1 rood.
4 roods.1 acre.
640 acres.1 square mile.

1 pound = 27.681 cubic inches distilled water at its maximum density (39.2° Far.).

Measures of Volume.

A cubic foot has	1728 cubic inches.
An ale gallon has	282 "
A standard or wine gallon has	231 "
A dry gallon has	268.8 "
A bushel has	2150.4 "
A cord of wood has	128 cubic feet.
A perch of stone has	24.75 "
A ton of round timber has	40 "
A ton of hewn timber has	50 "
A box 19⅝x19⅝ inches, 19⅝ inches deep, contains	1 barrel.
A box 12$\frac{11}{16}$x12$\frac{11}{16}$ inches, 12$\frac{11}{16}$ inches deep, contains	1 bushel.
A box 8½x8½ inches, 8½ inches deep, contains	1 peck.
A box 6$\frac{7}{16}$x6$\frac{7}{16}$ inches, 6$\frac{7}{16}$ inches deep, contains	½ "
A box 4$\frac{1}{16}$x$\frac{1}{16}$ inches, 4$\frac{1}{16}$ inches deep, contains	1 quart.
An acre contains	4840 square yards.
209 feet long by 209 feet broad is	1 acre.

Table of Distances.

A mile is	5280 feet or 1760 yards.
A league is	3 miles.
A fathom is	6 feet.
A metre is nearly	3 feet 3⅜ inches.
A hand is	4 inches.
A palm is	3 "
A span is	9 "

Liquid Measures.

A barrel holds.....31½ gallons.
A hogshead holds..63 "
A tierce holds.....42 "
A puncheon holds..84 "
A tun holds.....252 "

Barrel Measures.

A barrel of flour is 196 pounds.
A " " pork is 200 "
A " " rice is 600 "
A firkin of butter is 56 "
A tub of butter is 84 "

Measures of Length.

12 inches.....1 foot.	4 rods.....1 chain.
3 feet.....1 yard.	10 chains.....1 furlong.
2 yards.....1 fathom.	8 furlongs.....1 mile.
16½ feet.....1 rod.	3 miles.....1 league.

Nautical Measure.

A nautical (sea) mile or knot is the length of a minute of longitude of the earth at the equator and at sea level. It is assumed equal to 6086.07 feet, or, 1.152664 statute (land) miles by the United States Coast Survey. 3 nautical miles = 1 nautical league.

TABLE OF STANDARD DIMENSIONS OF WROUGHT IRON WELDED PIPES.

Nominal Diam.	*External Diam.*	*Thickness.*	*Internal Diam.*	*Internal Circum.*	*External Circum.*	*Length of Pipe per Sq. Foot of internal surface.*	*Length of Pipe per Sq. Foot of external surface.*	*Internal Area.*	*Weight per Foot.*	*No. of Threads per Inch of Screw.*
In.	In.	In.	In.	In.	In.	Feet	Feet	In.	Lbs.	
⅛	.40	.068	.27	.85	1.27	14.15	9.44	.057	.24	27
¼	.54	.088	.36	1.14	1.7	10.5	7.075	.104	.42	18
⅜	.67	.091	.49	1.55	2.12	7.67	5.657	.192	.56	18
½	.84	.109	.62	1.96	2.65	6.13	4.502	.305	.84	14
¾	1.05	.113	.82	2.59	3.3	4.64	3.637	.533	1.13	14
1	1.31	.134	1.05	3.29	4.13	3.66	2.903	.863	1.67	11½
1¼	1.66	.14	1.38	4.33	5.21	2.77	2.301	1.496	2.26	11½
1½	1.9	.145	1.61	5.06	5.97	2.37	2.01	2.038	2.69	11½
2	2.37	.154	2.07	6.49	7.46	1.85	1.611	3.355	3.67	11½
2½	2.87	.204	2.47	7.75	9.03	1.55	1.328	4.783	5.77	8
3	3.5	.217	3.07	9.64	11.	1.24	1.091	7.388	7.55	8
3½	4.	.226	3.55	11.15	12.57	1.08	0.955	9.887	9.05	8
4	4.5	.237	4.07	12.69	14.14	.95	0.849	12.73	10 73	8
4½	5.	.247	4.51	14.15	15.71	.85	0.765	15.939	12.49	8
5	5.56	.259	5.04	15.85	17.47	.78	0.629	19.99	14 56	8
6	6.62	.28	6.06	19.05	20.81	.63	0.577	28.889	18.77	8
7	7 62	.301	7.02	22.06	23.95	.54	0.505	38 737	23.41	8
8	8.62	.322	7.98	25.08	27.1	.48	0.444	50.039	28.35	8
9	9.69	.344	9.	28.28	30.45	.42	0.394	64.633	34.08	8
10	10 75	.366	10.02	31.47	33.77	.38	0.355	78.838	40.64	8

WEIGHT OF ROUND AND SQUARE ROLLED IRON.

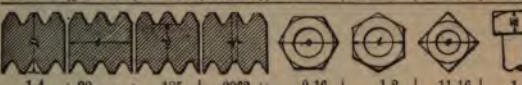
From $\frac{1}{16}$ Inch to 9$\frac{1}{8}$ Inches in Diameter and One Foot in Length.

Side or Diam. in Inches.	Weight in lbs. per foot.		Side or Diam. in Inches.	Weight in lbs. per foot.	
	Round.	Square.		Round.	Square.
1/16	.01	.013	3/8	39.864	50.756
1/8	.041	.053	4.	42.464	54.084
3/16	.093	.118	1/8	45.174	57.517
1/4	.165	.211	1/4	47.952	61.055
5/16	.373	.475	3/8	50.815	64.7
3/8	.663	.845	1/2	53.76	68.448
7/16	1.043	1.32	5/8	56.788	72.305
1/2	1.493	1.901	3/4	59.9	76.264
9/16	2.032	2.588	7/8	63.094	80.333
1.	2.654	3.38	5.	66.35	84.48
1/8	3.359	4.278	1/8	69.731	88.784
1/4	4.147	5.28	1/4	73.172	93.168
3/8	5.019	6.39	3/8	76.7	97.657
1/2	5.972	7.604	1/2	80.304	102.24
5/8	7.01	8.926	5/8	84.001	106.953
3/4	8.128	10.352	3/4	87.776	111.756
7/8	9.333	11.883	7/8	91.634	116.671
2.	10.616	13.52	6.	95.552	121.664
1/8	11.988	15.263	1/8	103.704	132.04
1/4	13.44	17.112	1/2	112.16	142.816
3/8	14.975	19.066	3/4	120.96	154.012
1/2	16.588	21.12	7.	130.048	165.632
5/8	18.293	23.292	1/4	139.544	177.672
3/4	20.076	25.56	1/2	149.328	190 136
7/8	21.944	27.939	3/4	159.456	203.024
3.	23.888	30.416	8.	169.856	216.336
1/8	25.926	33.01	1/4	180.696	230.068
1/4	28.04	35.704	1/2	191.808	244.22
3/8	30.24	38.503	3/4	203.26	258.8
1/2	32.512	41.408	9.	215.04	273.792
5/8	34.886	44.418	1/4	227.152	289.22
3/4	37.332	47.534	1/2	239.6	305.056

WEIGHT OF SHEET LEAD.

Thickness, Inches.	Weight, lbs.	Thickness, Inches.	Weight, lbs.	Thickness, Inches.	Weight, lbs.
.017	1	.085	5	.152	9
.034	2	.101	6	.169	10
.051	3	.118	7	.186	11
.068	4	.135	8	.203	12

PROPORTIONS OF THE UNITED STATES STANDARD SCREW-THREADS, NUTS AND BOLTHEADS.

Diam. of Screw.	Thre'ds per inch	Diam. of core.	Width of flat.	Outside diam.	Inside diam.	Diagonal.	Height of h'd.
							
1-4	20	.185	.0062	9-16	1-2	11-16	1-4
5-16	18	.240	.0070	11-16	19-32	13-16	19-64
3-8	16	.294	.0078	25-32	11-16	31-32	11-32
7-16	14	.344	.0089	49-48	25-32	1 1-16	25-64
1-2	13	.400	.0096	1	7-8	1 1-4	7-16
9-16	12	.454	.0104	1 7-64	31-32	1 5-16	31-64
5-8	11	.507	.0113	1 7-32	1 1-16	1 1-2	17-32
3-4	10	.620	.0125	1 7-16	1 1-4	1 3-4	5-8
7-8	9	.731	.0140	1 21-32	1 7-16	2 1-32	23-32
1	8	.837	.0156	1 7-8	1 5-8	2 5-16	13-16
1 1-8	7	.940	.0180	2 3-32	1 13-16	2 1-2	29-64
1 1-4	7	1.065	.0180	2 5-16	2	2 27-32	1
1 3-8	6	1.160	.0210	2 1-2	2 3-16	3 1-10	1 3-32
1 1-2	6	1.284	.0210	2 3-4	2 3-8	3 3-8	1 3-16
1 5-8	5 1-2	1.389	.0227	2 15-16	2 9-16	3 5-8	1 9-32
1 3-4	5	1.490	.0250	3 3-16	2 3-4	3 29-32	1 3-8
1 7-8	5	1.615	.0250	3 13-32	2 15 16	4 3-16	1 15-32
2	4 1-2	1.712	.0280	3 5-8	3 1-8	4 7-16	1 9-16
2 1-4	4 1-2	1.962	.0280	4 1-16	3 1-2	4 31-32	1 3-4
2 1-2	4	2.175	.0310	4 1-2	3 7-8	5 1-2	1 15-16
2 3-4	4	2.425	.0310	4 29-32	4 1-4	6	2 1-8
3	3 1-2	2.628	.0357	5 3-8	4 5-8	6 9-16	2 5-16
3 1-4	3 1-2	2.878	.0357	5 3-4	5	7 1-8	2 1-2
3 1-2	3 1-4	3.100	.0384	6 7-64	5 3-8	7 5-8	2 11-16
3 3-4	3	3.317	.0410	6 5-8	5 3-4	8 5-16	2 7-8
4	3	3.566	.0410	7 3-64	6 1-8	8 11-16	3 1-16
4 1-4	2 7-8	3.798	.0435	7 1-2	6 1-2	9 1-4	3 1-4
4 1-2	2 3-4	4.027	.0460	7 31-32	6 7-8	9 3-4	3 7-16
4 3-4	2 5-8	4.255	.0480	8 3-8	7 1-4	10 9-32	3 5-8
5	2 1-2	4.480	.0500	8 13-16	7 5-8	10 13-16	3 13-16
5 1-4	2 1-2	4.730	.0500	9 1-4	8	11 5-8	4
5 1-2	2 3-8	4.953	.0526	9 11-16	8 3-8	11 29-32	4 3-16
5 3-4	2 3-8	5.203	.0526	10 1-8	8 3-4	12 7-16	4 3-8
6	2 1-4	5.423	.0555	10 9-16	9 1-8	12 9-10	4 9-16

The threads have an angle of 60°, with flat tops and bottoms, and are of the following proportions:

Notation of letters. All dimensions in inches.

D = outside diameter of screw.
d = diameter of root of thread, or of hole in the nut.
p = pitch of screw.
t = number of threads per inch.
f = flat top and bottom.
o = outside diameter of hexagon nut or bolthead.
i = inside diameter of hexagon, or side of square nut or bolthead.
s = diagonal of square nut or bolthead.
h = height of rough or unfinished bolthead.

The height of finished nut or bolthead is made equal to the diameter D of the screw.

$$p = \frac{\sqrt{16\,D + 10} - 2.900}{16.64}. \qquad t = \frac{1}{p}. \qquad s = 1.414\,i.$$

$$d = D - \frac{1.299}{t}. \qquad i = \frac{3\,D}{2} + \frac{1}{8}. \qquad o = 1.155\,i. \qquad f = \frac{p}{8}.$$

SPECIFIC GRAVITY.

The specific gravity of a body is the ratio between its weight and the weight of a like volume of distilled water at a temperature of 39.2° Fahr. For aeriform bodies, air is taken as the unit.

Names of Substances.	*Specific gravity.*	*Weight per cub. inch.*
		lbs.
Metals.		
Platinum, rolled	22·669	·798
" wire	21·042	·761
" hammered	20·337	·736
Gold, hammered	19·361	·700
" pure cast	19·258	·697
" 22 carats fine	17·486	·733
Mercury, solid at—40° Fahr	15·632	·566
" at +32° Fahr	13·619	·493
" " 60° "	13·580	·491
" " 212° "	13·375	·484
Lead, pure	11·330	·410
" hammered	11·388	·412
Silver, hammered	10·511	·381
" pure	10·474	·379
Bismuth	9·823	·355
Copper, wire and rolled	8·878	·321
" pure	8·788	·318
Bronze, gun metal	8·700	·315
Brass, common	7·820	·282
Steel, cast steel	7·919	·286
" common soft	7·833	·283
" hardened and tempered	7·818	·283
Iron, pure	7·768	·281
" wrought and rolled	7·780	·282
" hammered	7·789	·282
" cast-iron	7·207	·261
Tin, from Böhmen	7·312	·265
" English	7·291	·264
Zinc, rolled	7·191	·260
Antimony	6·712	·244
Aluminum	2·500	·090
Stones and Earths.		
Emery	4·000	·144
Limestone	2·700	·097

SPECIFIC GRAVITY.—*Continued.*

Names of Substances.	*Specific gravity.*	*Weight per cub. inch.*
Asbestos, starry	3·073	·1110
Glass, flint	2·933	·1060
" white	2·892	·1040
" bottle	2·732	·0987
" green	2·642	·0954
Marble, Parian	2·838	·1030
" African	2·708	·0978
" Egyptian	2·668	·0964
Mica	2·800	·1000
Chalk	2·784	·1000
Coral, red	2·700	·0974
Granite, Susquehanna	2·704	·0976
" Quincy	2·652	·0958
" Patapsco	2·640	·0954
" Scotch	2·625	·0948
Marble, white Italian	2·708	·0978
" common	2·686	·0968
Talc, black	2·900	·0105
Quartz	2·660	·0962
Slate	2·672	·0965
Pearl, oriental	2·650	·0957
Shale	2·600	·0940
Flint, white	2·594	·0936
" black	2·582	·0933
Stone, common	2·520	·0910
" Bristol	2·510	·0906
" Mill	2·484	·0897
" Paving	2·416	·0873
Gypsum, opaque	2·168	·0783
Grindstone	2·143	·0775
Salt, common	2·130	·0770
Saltpetre	2·090	·0755
Sulphur, native	2·033	·0735
Common soil	1·984	·0717
Rotten stone	1·981	·0416
Clay	1·930	·0698
Brick	1·900	·0686
Nitre	1·900	·0686
Plaster Paris	1·872	·0677
	2·473	·0894
Ivory	1·822	·0659
Sand	2·650	·0571

SPECIFIC GRAVITY.—*Continued.*

Names of Substances.	*Specific gravity.*	*Weight per cub. inch.*
Phosphorus	1·770	·0640
Borax	1·714	·0620
Coal, Anthracite	1·640	·0593
	1·436	·0592
" Maryland	1·355	·0490
" Scotch	1·300	·0470
" Newcastle	1·270	·0460
" Bituminous	1·350	·0480
Earth, loose	1·500	·0542
Lime, quick	1·500	·0549
Charcoal	0·441	·0160
Woods (Dry).		
Alder	·800	·0289
Apple-tree	·793	·0287
Ash, the trunk	·845	·0306
Bay-tree	·822	·0297
Beech	·852	·0308
Box, French	·912	·0330
" Dutch	1·328	·0480
" Brazilian red	1·031	·0373
Cedar, wild	·596	·0219
" Palestine	·613	·0222
" American	·561	·0203
Cherry-tree	·715	·0259
Cork	·240	·0087
Ebony, American	1·331	·0481
Elder-tree	·695	·0252
Elm	·560	·0202
Filbert-tree	·600	·0217
Fir, male	·550	·0199
" female	·498	·0180
Hazel	·600	·0217
Lemon-tree	·703	·0254
Lignum-vitæ	1·333	·0482
Linden-tree	·604	·0219
Log-wood	·913	·0331
Mahogany, Honduras	·560	·0202
Maple	·750	·0271
Mulberry	·897	·0324
Oak	·950	·0343

ALLOYS.

A=Antimony, B=Bismuth, C=Copper, G=Gold, I=Iron, L=Lead, N=Nickel, S=Silver, T=Tin, Z=Zinc.

Name.	*Alloy.*
Brass, common yellow	$2C, 1Z$.
Brass, to be rolled	$32C, 10Z, 1.5T$.
Brass castings, common	$20C, 1.25Z, 2.5T$.
" " hard	$25C, 2Z, 4.5T$.
Brass propellers	$8C, 0.5Z, 1T$.
Gun-metal	$8C, 1T$.
Copper flanges	$9C, 1Z, 0.26T$.
Muntz's metal	$6C, 4Z$.
Statuary	$91.4C, 5.53Z, 1.7T, 1.37L$.
German Silver	$2C, 7.9N, 6.3Z, 6.5I$.
Britannia metal	$50A, 25T, 25B$.
Chinese Silver	$65.1C, 19.3Z, 13N, 2.58S, 12I$.
Chinese white Copper	$20.2C, 12.7Z, 1.3T, 15.8N$.
Medals	$100C, 8Z$.
Pinchbeck	$5C, 1Z$.
Babbitt's metal	$25T, 2A, 0.5C$.
Bell metal, large	$3C, 1T$.
" " small	$4C, 1T$.
Chinese gongs	$40.5C, 9.2T$.
Telescope mirrors	$33.3C, 16.7T$.
White metal, ordinary	$3.7C, 3.7Z, 14.2T, 28.4A$.
" " hard	$35C, 13Z, 2.2T$.
Sheeting metal	$56C, 45Z$, 12 arsenic.
Metal, expand in cooling	$75L, 16.7A, 8.3B$.

Imitation of Gold.

Melt separately	$\begin{cases} x=21C, 13T. \\ y=62C, 9Z. \end{cases}$
Gold imitation	$71y, 9x$.

Type Metals.

Name.	*Alloy.*
Smallest type	$3L, 1A$.
Small type	$4L, 1A$.
Medium type	$5L, 1A$.
Large type	$6L, 1A$.
Largest type	$7L, 1A$,
Metal which can be forged at red heat, and strong as good iron	$38.2Z, 60C, 1.75I$.

ALLOYS—*Continued.*

A—Antimony, *B*—Bismuth, *C*—Copper, *G*—Gold, *I*—Iron, *L*—Lead, *N*—Nickel, *S*—Silver, *T*—Tin, *Z*—Zinc.

Alloys for Solders.

Name.	*Alloy.*	*Melts.*
Newton's fusible	8*B*, 5*L*, 3*T*,	212°
Rose's "	2*B*, 1*L*, 1*T*,	201°
A more "	5*B*, 3*L*, 2*T*,	199°
Still more "	12*T*, 25*L*, 50*B*, 13 cadmium,	155°
For tin solder, coarse	1*T*, 3*L*,	500°
For tin solder, ordinary	2*T*, 1*L*,	360°
For brass, soft spelter	1*C*, 1*Z*,	550°
Hard, for iron	2*C*, 1*Z*,	790°
For steel	19*S*, 3*C*, 1*Z*.	
For fine brass work	1*S*, 8*C*, 8*Z*.	
Pewterer's soft solder	2*B*, 4*L*, 3*T*.	
Pewterer's soft solder	1*B*, 1*L*, 2*T*.	
Gold solder	24*G*, 2*S*, 1*C*.	
Silver solder, hard	4*S*, 1*C*.	
" " soft	2*S*, 1 brass wire.	
For Lead	16*T*, 33*L*.	

Fluxes for Soldering or Welding.

Iron	Borax	Zinc	Chloride of Zinc
Tinned Iron	Resin	Lead	Tallow or Resin
Copper & Brass	Sal-ammoniac	Lead and Tin Pipes	Resin and Sweet Oil

Steel.—Pulverize together one part of sal-ammoniac and ten parts of borax and fuse until clear; When solidified, pulverize to powder.

TEMPERING OF STEEL.

The following colors may be made use of in tempering steel cutting tools:

		Corresponding Temperature, Fahr.
Lancets	Pale yellow	430°
Razors	Straw yellow	450°
All kinds of wood-cutting tools	Darker straw yellow	470°
Screw taps	Yellow	490°
Chipping chisels, hatchets and saws	Brown yellow	500°
All kinds of percussive tools	Brown (slightly tinged purple)	520°
	Light purple	530°
Springs	Clear black	570°
	Dark blue	600°

CIRCUMFERENCES AND AREAS OF CIRCLES FROM 1/64 TO 100.

Diam.	Circum.	Area.	Diam.	Circum.	Area.
1/64	.04909	.000192	3/8	13.7445	15.033
1/32	.09818	.000767	1/2	14.1372	15.9043
1/16	.19635	.003068	5/8	14.5299	16.8002
3/32	.3927	.012272	3/4	14.9226	17.7206
1/8	.589	.027612	7/8	15.3153	18.6555
3/16	.7854	.049087	5.	15.708	19.635
1/4	.98175	.076699	1/8	16.1007	20.629
5/16	1.1781	.110447	1/4	16.4934	21.6476
3/8	1.37445	.15032	3/8	16.8861	22.6907
7/16	1.5708	.19635	1/2	17.2788	23.7583
1/2	1.76715	.248505	5/8	17.6715	24.8505
9/16	1.9635	.306796	3/4	18.0642	25.9673
5/8	2.15985	.371224	7/8	18.4569	27.1086
11/16	2.3562	.441787	6.	18.8496	28.2744
3/4	2.55255	.518487	1/8	19.2423	29.4648
13/16	2.7489	.601322	1/4	19.635	30.6797
7/8	2.94525	.690292	3/8	20.0277	31.9191
15/16	3.1416	.7854	1/2	20.4204	33.1831
1.	3 5343	.99402	5/8	20.8131	34.4717
1/16	3.927	1.2272	3/4	21.2058	35.7848
1/8	4.3197	1.4849	7/8	21.5985	37.1224
3/16	4.7124	1.7671	7.	21.9912	38.4846
1/4	5.1051	2.0739	1/8	22.3839	39.8713
5/16	5.4978	2.4053	1/4	22.7766	41.2826
3/8	5.8905	2.7612	3/8	23.1693	42.7184
7/16	6.2832	3.1416	1/2	23.562	44.1787
1/2	6.6759	3.5466	5/8	23.9547	45.6636
9/16	7.0686	3.9761	3/4	24.3474	47.1731
5/8	7.4613	4.4301	7/8	24.7401	48.7071
11/16	7.854	4.9087	8.	25.1328	50.2656
3/4	8.2467	5.4119	1/8	25.5255	51.8487
13/16	8.6394	5.9396	1/4	25.9182	53 4563
7/8	9.0321	6.4918	3/8	26.3109	55.0884
15/16	9.4248	7.0686	1/2	26.7036	56.7451
1/16	9.8175	7.6699	5/8	27.0963	58.4264
1/8	10.2102	8.2958	3/4	27.489	60.1322
3/16	10.6029	8.9462	7/8	27.8817	61.8625
1/4	10.9956	9.6211	9.	28.2744	63.6174
5/16	11.3883	10.3206	1/8	28.6671	65.3968
3/8	11.781	11.0447	1/4	29.0598	67.2008
7/16	12.1737	11.7933	3/8	29.4525	69.0293
4.	12.5664	12.5664	1/2	29.8452	70.8823
1/16	12.9591	13.3641	5/8	30.2379	72.7599
1/8	13.3518	14.1863	3/4	30.6306	74.6621

TABLE — *Continued.*

m.	Circum.	Area.	Diam.	Circum.	Area.
	31.0233	76.5888	.3/8	48.3021	185.661
	31.416	78.54	.1/2	48.6948	188.692
	31.8087	80.5158	.5/8	49.0875	191.748
	32.2014	82.5161	.3/4	49.4802	194.828
	32.5941	84.5409	.7/8	49.8729	197.933
	32.9868	86.5903	16.	50.2656	201.062
	33.3795	88.6643	.1/8	50.6583	204.216
	33.7722	90.7628	.1/4	51.051	207.395
	34.1649	92.8858	.3/8	51.4437	210.598
	34.5576	95.0334	.1/2	51.8364	213.825
	34.9503	97.2055	.5/8	52.2291	217.077
	35.343	99.4022	.3/4	52.6218	220.354
	35.7357	101 6234	.7/8	53.0145	223.655
	36.1284	103.8691	17.	53.4072	226.981
	36.5211	106.1394	.1/8	53.7999	230.331
	36.9138	108.4343	.1/4	54.1926	233.706
	37.3065	110.7537	.3/8	54.5853	237.105
	37.6992	113.098	.1/2	54.978	240.529
	38.0919	115.466	.5/8	55.3707	243.977
	38.4846	117.859	.3/4	55.7634	247.45
	38.8773	120.277	.7/8	56.1561	250.948
	39.27	122.719	18.	56.5488	254.47
	39.6627	125.185	.1/8	56.9415	258.016
	40.0554	127.677	.1/4	57.3342	261.587
	40.4481	130.192	.3/8	57.7269	265.183
	40.8408	132.733	.1/2	58.1196	268.803
	41.2335	135.297	.5/8	58.5123	272.448
	41.6262	137.887	.3/4	58.905	276.117
	42.0189	140.501	.7/8	59.2977	279.811
	42.4116	143.139	19.	59.6904	283.529
	42.8043	145.802	.1/8	60.0831	287.272
	43.197	148.49	.1/4	60.4758	291.04
	43.5897	151.202	.3/8	60.8685	294.832
	43.9824	153.938	.1/2	61.2612	298.648
	44.3751	156.7	.5/8	61.6539	302.489
	44.7678	159.485	.3/4	62.0466	306.355
	45.1605	162.296	.7/8	62.4393	310.245
	45.5532	165.13	20.	62.832	314.16
	45.9459	167.99	.1/8	63.2247	318.099
	46.3386	170.874	.1/4	63.6174	322.063
	46.7313	173.782	.3/8	64.0101	326.051
	47.124	176.715	.1/2	64.4028	330.064
	47.5167	179.673	.5/8	64.7955	334.102
	47.9094	182.655	.3/4	65.1882	338.164

TABLE—*Continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.
⅞	65.5809	342.25	⅜	82.8597	546.356
21.	65.9736	346.361	½	83.2524	551,547
⅛	66.3663	350.497	⅝	83.6451	556.763
¼	66.759	354.657	¾	84.0378	562.003
⅜	67.1517	358.842	⅞	84.4305	567.267
½	67.5444	363.051	27.	84.8232	572.557
⅝	67.9371	367.285	⅛	85.2159	577.87
¾	68.3298	371.543	¼	85.6086	583.209
⅞	68.7225	375.826	⅜	86.0013	588.571
22.	69.1152	380.134	½	86.394	593.959
⅛	69.5079	384.466	⅝	86.7867	599.371
¼	69.9006	388.822	¾	87.1794	604.807
⅜	70.2933	393 203	⅞	87.5721	610.268
½	70.686	397.609	28.	87.9648	615.754
⅝	71.0787	402.038	⅛	88.3575	621.264
¾	71.4714	406.494	¼	88.7502	626.798
⅞	71.8641	410.973	⅜	89.1429	632.357
23.	72.2568	415.477	½	89.5356	637.941
⅛	72.6495	420.004	⅝	89.9283	643.549
¼	73.0422	424.558	¾	90.321	649.182
⅜	73.4349	429.135	⅞	90.7137	654.84
½	73.8276	433.737	29.	31.1064	660.521
⅝	74.2203	438.364	⅛	91.4991	666.228
¾	74.613	443.015	¼	91.8918	671.959
⅞	75.0057	447.69	⅜	92.2845	677.714
24.	75.3984	452.39	½	92.6772	683.494
⅛	75.7911	457.115	⅝	93.0699	689.299
¼	76.1838	461.864	¾	93.4626	695.128
⅜	76.5765	466.638	⅞	93.8553	700.982
½	76.9692	471.436	30.	94.248	706.86
⅝	77.3619	476.259	⅛	94.6407	712.763
¾	77.7546	481.107	¼	95.0334	718.69
⅞	78.1473	485.979	⅜	95.4261	724.642
25.	78.54	490.875	½	95.8188	730.618
⅛	78.9327	495.796	⅝	96.2115	736.619
¼	79.3254	500.742	¾	96.6042	742.645
⅜	79.7181	505.712	⅞	96.9969	748.695
½	80.1108	510.706	31.	97.3896	754.769
⅝	80.5035	515.726	⅛	97.7823	760.869
¾	80.8962	520.769	¼	98.175	766.992
⅞	81.2889	525.838	⅜	98.5677	773.14
26.	81.6816	530.93	½	98.9604	779.313
⅛	82.0743	536.048	⅝	99.3531	785.51
¼	82.467	541.19	¾	99.7458	791.732

TABLE — *Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
.⅜	100.1385	797.979	.⅝	117.417	1097.118
32.	100.5312	804.25	.¾	117.81	1104.469
.⅛	100.9239	810.545	.⅞	118.203	1111.844
.¼	101.3166	816.865	.¾	118.595	1119.244
.⅜	101.7093	823.21	.⅞	118.988	1126.669
.½	102.102	829.579	38.	119.381	1134.118
.⅝	102.4947	835.972	.⅛	119.773	1141.591
.¾	102.8874	842.391	.¼	120.166	1149.089
.⅞	103.2801	848.833	.⅜	120.559	1156.612
33.	103.673	855.301	.½	120.952	1164.159
.⅛	104.065	861.792	.⅝	121.344	1171.731
.¼	104.458	868.309	.¾	121.737	1179.327
.⅜	104.851	874.85	.⅞	122.13	1186.948
.½	105.244	881.415	39.	122.522	1194.593
.⅝	105.636	888.005	.⅛	122.915	1202.263
.¾	106.029	894.62	.¼	123.308	1209.958
.⅞	106.422	901.259	.⅜	123.7	1217.677
34.	106.814	907.922	.½	124.093	1225.42
.⅛	107.207	914.611	.⅝	124.486	1233.188
.¼	107.6	921.323	.¾	124.879	1240.981
.⅜	107.992	928.061	.⅞	125.271	1248.798
.½	108.385	934.822	40.	125.664	1256.64
.⅝	108.778	941.609	.⅛	126.057	1264.51
.¾	109.171	948.42	.¼	126.449	1272.4
.⅞	109.563	955.255	.⅜	126.842	1280.31
35.	109.956	962.115	.½	127.235	1288.25
.⅛	110.349	969.	.⅝	127.627	1296.22
.¼	110.741	975.909	.¾	128.02	1304.21
.⅜	111.134	982.842	.⅞	128.413	1312.22
.½	111.527	989.8	41.	128.806	1320.26
.⅝	111.919	996.783	.⅛	129.198	1328.32
.¾	112.312	1003.79	.¼	129.591	1336.41
.⅞	112.705	1010.822	.⅜	129.984	1344.52
36.	113.098	1017.878	.½	130.376	1352.66
.⅛	113.49	1024.96	.⅝	130.769	1360.82
.¼	113.883	1032.065	.¾	131.162	1369.
.⅜	114.276	1039.195	.⅞	131.554	1377.21
.½	114.668	1046.349	42.	131.947	1385.45
.⅝	115.061	1053.528	.⅛	132.34	1393.7
.¾	115.454	1060.732	.¼	132.733	1401.99
.⅞	115.846	1067.96	.⅜	133.125	1410.3
37.	116.239	1075.213	.½	133.518	1418.63
.⅛	116.632	1082.49	.⅝	133.911	1426.99
.¼	117.025	1089.792	.¾	134.303	1435.37

TABLE—*Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅞	65.5809	342.25	⅝	82.8597	546.356
21.	65.9736	346.361	¾	83.2524	551.547
⅛	66.3663	350.497	⅞	83.6451	556.763
¼	66.759	354.657	26.	84.0378	562.003
⅜	67.1517	358.842	⅛	84.4305	567.267
½	67.5444	363.051	¼	84.8232	572.557
⅝	67.9371	367.285	⅜	85.2159	577.87
¾	68.3298	371.543	½	85.6086	583.209
⅞	68.7225	375.826	⅝	86.0013	588.571
22.	69.1152	380.134	¾	86.394	593.959
⅛	69.5079	384.466	⅞	86.7867	599.371
¼	69.9006	388.822	27.	87.1794	604.807
⅜	70.2933	393 203	⅛	87.5721	610.268
½	70.686	397.609	¼	87.9648	615.754
⅝	71.0787	402.038	⅜	88.3575	621.264
¾	71.4714	406.494	½	88.7502	626.798
⅞	71.8641	410.973	⅝	89.1429	632.357
23.	72.2568	415.477	¾	89.5356	637.941
⅛	72.6495	420.004	⅞	89.9283	643.549
¼	73.0422	424.558	28.	90.321	649.182
⅜	73.4349	429.135	⅛	90.7137	654.84
½	73.8276	433.737	¼	31.1064	660.521
⅝	74.2203	438.364	⅜	91.4991	666.228
¾	74.613	443.015	½	91.8918	671.959
⅞	75.0057	447.69	⅝	92.2845	677.714
24.	75.3984	452.39	¾	92.6772	683.494
⅛	75.7911	457.115	⅞	93.0699	689.299
¼	76.1838	461.864	29.	93.4626	695.128
⅜	76.5765	466.638	⅛	93.8553	700.982
½	76.9692	471.436	¼	94.248	706.86
⅝	77.3619	476.259	⅜	94.6407	712.763
¾	77.7546	481.107	½	95.0334	718.69
⅞	78.1473	485.979	⅝	95.4261	724.642
25.	78.54	490.875	¾	95.8188	730.618
⅛	78.9327	495.796	⅞	96.2115	736.619
¼	79.3254	500.742	30.	96.6042	742.645
⅜	79.7181	505.712	⅛	96.9969	748.695
½	80.1108	510.706	¼	97.3896	754.769
⅝	80.5035	515.726	⅜	97.7823	760.869
¾	80.8962	520.769	½	98.175	766.992
⅞	81.2889	525.838	⅝	98.5677	773.14
26.	81.6816	530.93	¾	98.9604	779.313
⅛	82.0743	536.048	⅞	99.3531	785.51
¼	82.467	541.19	31.	90.7458	791.732

TABLE — *Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅞	100.1385	797.979	⅜	117.417	1097.118
32.	100.5312	804.25	½	117.81	1104.469
⅛	100.9239	810.545	⅝	118.203	1111.844
¼	101.3166	816.865	¾	118.595	1119.244
⅜	101.7093	823.21	⅞	118.988	1126.669
½	102.102	829.579	38.	119.381	1134.118
⅝	102.4947	835.972	⅛	119.773	1141.591
¾	102.8874	842.391	¼	120.166	1149.089
⅞	103.2801	848.833	⅜	120.559	1156.612
33.	103.673	855.301	½	120.952	1164.159
⅛	104.065	861.792	⅝	121.344	1171.731
¼	104.458	868.309	¾	121.737	1179.327
⅜	104.851	874.85	⅞	122.13	1186.948
½	105.244	881.415	39.	122.522	1194.593
⅝	105.636	888.005	⅛	122.915	1202.263
¾	106.029	894.62	¼	123.308	1209.958
⅞	106.422	901.259	⅜	123.7	1217.677
34.	106.814	907.922	½	124.093	1225.42
⅛	107.207	914.611	⅝	124.486	1233.188
¼	107.6	921.323	¾	124.879	1240.981
⅜	107.992	928.061	⅞	125.271	1248.798
½	108.385	934.822	40.	125.664	1256.64
⅝	108.778	941.609	⅛	126.057	1264.51
¾	109.171	948.42	¼	126.449	1272.4
⅞	109.563	955.255	⅜	126.842	1280.31
35.	109.956	962.115	½	127.235	1288.25
⅛	110.349	969.	⅝	127.627	1296.22
¼	110.741	975.909	¾	128.02	1304.21
⅜	111.134	982.842	⅞	128.413	1312.22
½	111.527	989.8	41.	128.806	1320.26
⅝	111.919	996.783	⅛	129.198	1328.32
¾	112.312	1003.79	¼	129.591	1336.41
⅞	112.705	1010.822	⅜	129.984	1344.52
36.	113.098	1017.878	½	130.376	1352.66
⅛	113.49	1024.96	⅝	130.769	1360.82
¼	113.883	1032.065	¾	131.162	1369.
⅜	114.276	1039.195	⅞	131.554	1377.21
½	114.668	1046.349	42.	131.947	1385.45
⅝	115.061	1053.528	⅛	132.34	1393.7
¾	115.454	1060.732	¼	132.733	1401.99
⅞	115.846	1067.96	⅜	133.125	1410.3
37.	116.239	1075.213	½	133.518	1418.63
⅛	116.632	1082.49	⅝	133.911	1426.99
¼	117.025	1089.792	¾	134.303	1435.37

TABLE—*Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅞	134.696	1443.77	⅜	151.975	1837.95
43.	135.089	1452.2	½	152.368	1847.46
⅛	135.481	1460.66	⅝	152.76	1856.99
¼	135.874	1469.14	¾	153.153	1866.55
⅜	136.267	1477.64	⅞	153.546	1876.14
½	136.66	1486.17	49.	153.938	1885.75
⅝	137.052	1494.73	⅛	154.331	1895.38
¾	137.445	1503.3	¼	154.724	1905.04
⅞	137.838	1511.91	⅜	155.116	1914.72
44.	138.23	1520.53	½	155.509	1924.43
⅛	138.623	1529.19	⅝	155.902	1934.16
¼	139.016	1537.86	¾	156.295	1943.91
⅜	139.408	1546.56	⅞	156.687	1953.69
½	139.801	1555.29	50.	157.08	1963.5
⅝	140.194	1564.04	⅛	157.473	1973.33
¾	140.587	1572.81	¼	157.865	1983.18
⅞	140.979	1581.61	⅜	158.258	1993.06
45.	141.372	1590.43	½	158.651	2002.97
⅛	141.765	1599.28	⅝	159.043	2012.89
¼	142.157	1608.16	¾	159.436	2022.85
⅜	142.55	1617.05	⅞	159.829	2032.82
½	142.943	1625.97	51.	160.222	2042.83
⅝	143.335	1634.92	⅛	160.614	2052.85
¾	143.728	1643.89	¼	161.007	2062.9
⅞	144.121	1652.89	⅜	161.4	2072.98
46.	144.514	1661.91	½	161.792	2083.08
⅛	144.906	1670.95	⅝	162.185	2093.2
¼	145.299	1680.02	¾	162.578	2103.35
⅜	145.692	1689.11	⅞	162.97	2113.52
½	146.084	1698.23	52.	163.363	2123.72
⅝	146.477	1707.37	⅛	163.756	2133.94
¾	146.87	1716.54	¼	164.149	2144.19
⅞	147.262	1725.73	⅜	164.541	2154.46
47.	147.655	1734.95	½	164.934	2164.76
⅛	148.048	1744.19	⅝	165.327	2175.08
¼	148.441	1753.45	¾	165.719	2185.42
⅜	148.833	1762.74	⅞	166.112	2195.79
½	149.226	1772.06	53.	166.505	2206.19
⅝	149.619	1781.4	⅛	166.897	2216.61
¾	150.011	1790.76	¼	167.29	2227.05
⅞	150.404	1800.15	⅜	167.683	2237.52
48.	150.797	1809.56	½	168.076	2248.01
⅛	151.189	1819	⅝	168.468	2258.53
¼	151.582	1828.46	¾	168.861	2269.07

TABLE — *Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
.⅜	169.254	2279.64	.⅜	186.532	2768.84
54.	169.646	2290.23	.½	186.925	2780.51
.⅛	170.089	2300.84	.⅝	187.318	2792.21
.¼	170.432	2311.48	.¾	187.711	2803.93
.⅜	170.824	2322.15	.⅞	188.103	2815.67
.½	171.217	2332.83	60.	188.496	2827.44
.⅝	171.61	2343.55	.⅛	188.889	2839.23
.¾	172.003	2354.29	.¼	189.281	2851.05
.⅞	172.395	2365.05	.⅜	189.674	2862.89
55.	172.788	2375.83	.½	190.067	2874.76
.⅛	173.181	2386.65	.⅝	190.459	2886.65
.¼	173.573	2397.48	.¾	190.852	2898.57
.⅜	173.966	2408.34	.⅞	191.245	2910.51
.½	174.359	2419.23	61.	191.638	2922.47
.⅝	174.751	2430.14	.⅛	192.03	2934.46
.¾	175.144	2441.07	.¼	192.423	2946.48
.⅞	175.537	2452.03	.⅜	192.816	2958.52
56.	175.93	2463.01	.½	193.208	2970.58
.⅛	176.322	2474.02	.⅝	193.601	2982.67
.¼	176.715	2485.05	.¾	193.994	2994.78
.⅜	177.108	2496.11	.⅞	194.386	3006.92
.½	177.5	2507.19	62.	194.779	3019.08
.⅝	177.893	2518.3	.⅛	195.172	3031.26
.¾	178.286	2529.43	.¼	195.565	3043.47
.⅞	178.678	2540.58	.⅜	195.957	3055.71
57.	179.071	2551.76	.½	196.35	3067.97
.⅛	179.464	2562.97	.⅝	196.743	3080.25
.¼	179.857	2574.2	.¾	197.135	3092.56
.⅜	180.249	2585.45	.⅞	197.528	3104.89
.½	180.642	2596.73	63.	197.921	3117.25
.⅝	181.035	2608.03	.⅛	198.313	3129.64
.¾	181.427	2619.36	.¼	198.706	3142.04
.⅞	181.82	2630.71	.⅜	199.099	3154.47
58.	182.213	2642.09	.½	199.492	3166.93
.⅛	182.605	2653.49	.⅝	199.884	3179.41
.¼	182.998	2664.91	.¾	200.277	3191.91
.⅜	183.391	2676.36	.⅞	200.67	3204.44
.½	183.784	2687.84	64.	201.062	3217.
.⅝	184.176	2699.33	.⅛	201.455	3229.58
.¾	184.569	2710.86	.¼	201.848	3242.18
.⅞	184.962	2722.41	.⅜	202.24	3254.81
59.	185.354	2733.98	.½	202.633	3267.46
.⅛	185.747	2745.57	.⅝	203.026	3280.14
.¼	186.14	2757.2	.¾	203.419	3292.84

TABLE—*Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅜	203.811	3305.56	⅜	221.09	3889.8
65.	204.204	3318.31	½	221.483	3903.63
⅛	204.597	3331.09	⅝	221.875	3917.49
¼	204.989	3343.89	¾	222.268	3931.37
⅜	205.382	3356.71	⅞	222.661	3945.27
½	205.775	3369.56	71.	223.054	3959.2
⅝	206.167	3382.44	⅛	223.446	3973.15
¾	206.56	3395.33	¼	223.839	3987.13
⅞	206.953	3408.26	⅜	224.232	4001.13
66.	207.346	3421.2	½	224.624	4015.16
⅛	207.738	3434.17	⅝	225.017	4029.21
¼	208.131	3447.17	¾	225.41	4043.29
⅜	208.524	3460.19	⅞	225.802	4057.39
½	208.916	3473.24	72.	226.195	4071.51
⅝	209.309	3486.3	⅛	226.588	4085.66
¾	209.702	3499.4	¼	226.981	4099.84
⅞	210.094	3512.52	⅜	227.373	4114.04
67.	210.487	3525.66	½	227.766	4128.26
⅛	210.88	3538.83	⅝	228.159	4142.51
¼	211.273	3552.02	¾	228.551	4156.78
⅜	211.665	3565.24	⅞	228.944	4171.08
½	212.058	3578.48	73.	229.337	4185.4
⅝	212.451	3591.74	⅛	229.729	4199.74
¾	212.843	3605.04	¼	230.122	4214.11
⅞	213.236	3618.35	⅜	230.515	4228.51
68.	213.629	3631.69	½	230.908	4242.93
⅛	214.021	3645.05	⅝	231.3	4257.37
¼	214.414	3658.44	¾	231.693	4271.84
⅜	214.807	3671.86	⅞	232.086	4286.33
½	215.2	3685.29	74.	232.478	4300.85
⅝	215.592	3698.76	⅛	232.871	4315.39
¾	215.985	3712.24	¼	233.264	4329.96
⅞	216.378	3725.75	⅜	233.656	4344.55
69.	216.77	3739.29	½	234.049	4359.17
⅛	217.163	3752.85	⅝	234.442	4373.81
¼	217.556	3766.43	¾	234.835	4388.47
⅜	217.948	3780.04	⅞	235.227	4403.16
½	218.341	3793.68	75.	235.62	4417.87
⅝	218.734	3807.34	⅛	236.013	4432.61
¾	219.127	3821.02	¼	236.405	4447.38
⅞	219.519	3834.73	⅜	236.798	4462.16
70.	219.912	3848.46	½	237.191	4476.98
⅛	220.305	3862.22	⅝	237.583	4491.81
¼	220.697	3876.	¾	237.976	4506.67

TABLE *Continued.*

Diam.	Circum.	Area.	Diam.	Circum.	Area.
.⅜	238.369	4521.56	.⅝	255.648	5200.83
76.	238.762	4536.47	.¹⁄₂	256.04	5216.82
.⅛	239.154	4551.41	.⅝	256.433	5232.84
.¼	239.547	4566.36	.¾	256.826	5248.88
.⅜	239.94	4581.35	.⅞	257.218	5264.94
.½	240.332	4596.36	82.	257.611	5281.03
.⅝	240.725	4611.39	.⅛	258.004	5297.14
.¾	241.118	4626.45	.¼	258.397	5313.28
.⅞	241.51	4641.53	.⅜	258.789	5329.44
77.	241.903	4656.64	.½	259.182	5345.63
.⅛	242.296	4671.77	.⅝	259.575	5361.84
.¼	242.689	4686.92	.¾	259.967	5378.08
.⅜	243.081	4702.1	.⅞	260.36	5394.34
.½	243.474	4717.31	83.	250.753	5410.62
.⅝	243.867	4732.54	.⅛	261.145	5426.93
.¾	244.259	4747.79	.¼	261.538	5443.26
.⅞	244.652	4763.07	.⅜	261.931	5459.62
78.	245.045	4778.37	.½	262.324	5476.01
.⅛	245.437	4793.7	.⅝	262.716	5492.41
.¼	245.83	4809.05	.¾	263.109	5508.84
.⅜	246.223	4824.43	.⅞	263.502	5525.3
.½	246.616	4839.83	84.	263.894	5541.78
.⅝	247.008	4855.26	.⅛	264.287	5558.29
.¾	247.401	4870.71	.¼	264.68	5574.82
.⅞	247.794	4886.18	.⅜	265.072	5591.37
79.	248.186	4901.68	.½	265.465	5607.95
.⅛	248.579	4917.21	.⅝	265.858	5624.56
.¼	248.972	4932.75	.¾	266.251	5641.18
.⅜	249.364	4948.33	.⅞	266.643	5657.84
.½	249.757	4963.92	85.	267.036	5674.51
.⅝	250.15	4979.55	.⅛	267.429	5691.22
.¾	250.543	4995.19	.¼	267.821	5707.94
.⅞	250.935	5010.86	.⅜	268.214	5724.69
80.	251.328	5026.56	.½	268.607	5741.47
.⅛	251.721	5042.28	.⅝	268.999	5758.27
.¼	252.113	5058.03	.¾	269.392	5775.1
.⅜	252.506	5073.79	.⅞	269.785	5791.94
.½	252.899	5089.59	86.	270.178	5808.82
.⅝	253.291	5105.41	.⅛	270.57	5825.72
.¾	253.684	5121.25	.¼	270.963	5842.64
.⅞	254.077	5137.12	.⅜	271.356	5859.59
81.	254.47	5153.01	.½	271.748	5876.56
.⅛	254.862	5168.93	.⅝	272.141	5893.55
.¼	255.255	5184.87	.¾	272.534	5910.58

TABLE — *Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅜	272.926	5927.62	⅜	290.205	6701.93
87.	273.319	5944.69	½	290.598	6720.08
⅛	273.712	5961.79	⅝	290.991	6738.25
¼	274.105	5978.91	¾	291.383	6756.45
⅜	274.497	5996.05	⅞	291.776	6774.68
½	274.89	6013.22	93.	292.169	6792.92
⅝	275.283	6030.41	⅛	292.562	6811.2
¾	275.675	6047.63	¼	292.954	6829.49
⅞	276.068	6064.87	⅜	293.347	6847.82
88.	276.461	6082.14	½	293.74	6866.16
⅛	276.853	6099.43	⅝	294.132	6884.53
¼	277.246	6116.74	¾	294.525	6902.93
⅜	277.629	6134.08	⅞	294.918	6921.35
½	278.032	6151.45	94.	295.31	6939.79
⅝	278.424	6168.84	⅛	295.703	6958.26
¾	278.817	6186.25	¼	296.096	6976.76
⅞	279.21	6203.69	⅜	296.488	6995.28
89.	279.602	6221.15	½	296.881	7013.82
⅛	279.995	6238.64	⅝	297.274	7032.39
¼	280.388	6256.15	¾	297.667	7050.98
⅜	280.78	6273.69	⅞	298.059	7069.59
½	281.173	6291.25	95.	298.452	7088.24
⅝	281.566	6308.84	⅛	298.845	7106.9
¾	281.959	6326.45	¼	299.237	7125.59
⅞	282.351	6344.08	⅜	299.63	7144.31
90.	282.744	6361.74	½	300.023	7163.04
⅛	283.137	6379.42	⅝	300.415	7181.81
¼	283.529	6397.13	¾	300.808	7200.6
⅜	283.922	6414.86	⅞	301.201	7219.41
½	284.315	6432.62	96.	301.594	7238.25
⅝	284.707	6450.4	⅛	301.986	7257.11
¾	285.1	6468.21	¼	302.379	7275.99
⅞	285.493	6486.04	⅜	302.772	7294.91
91.	285.886	6503.9	½	303.164	7313.84
⅛	286.278	6521.78	⅝	303.557	7332.8
¼	286.671	6539.68	¾	303.95	7351.79
⅜	287.064	6557.61	⅞	304.342	7370.79
½	287.456	6575.56	97.	304.735	7389.83
⅝	287.849	6593.54	⅛	305.128	7408.89
¾	288.242	6611.55	¼	305.521	7427.97
⅞	288.634	6629.57	⅜	305.913	7447.08
92.	289.027	6647.63	½	306.306	7466.21
⅛	289.42	6665.7	⅝	306.699	7485.37
¼	289.813	6683.8	¾	307.091	7504.55

TABLE — *Continued.*

Diam.	*Circum.*	*Area.*	*Diam.*	*Circum.*	*Area.*
⅞	307.484	7523.75	99.	311.018	7697.71
98.	307.877	7542.98	⅛	311.411	7717.16
⅛	308.27	7562.24	¼	311.804	7736.63
¼	308.662	7581.52	⅜	312.196	7756.13
⅜	309.055	7600.82	½	312.589	7775.66
½	309.448	7620.15	⅝	312.982	7795.21
⅝	309.84	7639.5	¾	313.375	7814.78
¾	310.233	7658.88	⅞	313.767	7834.38
⅞	310.626	7678.28	100.	314.16	7854.

TABLE OF DECIMAL EQUIVALENTS OF PARTS OF ONE INCH.

1-64	.015625	17-64	.265625	33-64	.515625	49-64	.765625
1-32	.03125	9-32	.28125	17-32	.53125	25-32	.78125
3-64	.046875	19-64	.296875	35-64	.546875	51-64	.796875
1-16	.0625	**5-16**	.3125	**9-16**	.5625	**13-16**	.8125
5-64	.078125	21-64	.328125	37-64	.578125	53-64	.828125
3-32	.09375	11-32	.34375	19-32	.59375	27-32	.84375
7-64	.109375	23-64	.359375	39-64	.609375	55-64	.859375
1-8	.125	**3-8**	.375	**5-8**	.625	**7-8**	.875
9-64	.140625	25-64	.390625	41-64	.640625	57-64	.890625
5-32	.15625	13-32	.40625	21-32	.65625	29-32	.90625
11-64	.171875	27-64	.421875	43-64	.671875	59-64	.921875
3-16	.1875	**7-16**	.4375	**11-16**	.6875	**15-16**	.9375
13-64	.203125	29-64	.453125	45-64	.703125	61-64	.953125
7-32	.21875	15-32	.46875	23-32	.71875	31-32	.96875
15-64	.234375	31-64	.484375	47-64	.734375	63-64	.984375
1-4	.25	**1-2**	.50	**3-4**	.75	**1**	

DECIMALS OF A FOOT FOR EACH $\frac{1}{32}$ OF AN INCH.

Inch.	0"	1"	2"	3"	4"	5"
0	0	.0833	.1667	.2500	.3333	.4167
$\frac{1}{32}$	.0026	.0859	.1693	.2526	.3359	.4193
$\frac{1}{16}$	.0052	.0885	.1719	.2552	.3385	.4219
$\frac{3}{32}$	.0078	.0911	.1745	.2578	.3411	.4245
$\frac{1}{8}$	.0104	.0937	.1771	.2604	.3437	.4271
$\frac{5}{32}$	.0130	.0964	.1797	.2630	.3464	.4297

TABLE — *Continued.*

Inch.	0"	1"	2"	3"	4"	5"
1/64	.0156	.0990	.1823	.2656	.3490	.4323
1/32	.0182	.1016	.1849	.2682	.3516	.4349
1/16	.0208	.1042	.1875	.2708	.3542	.4357
3/64	.0234	.1068	.1901	.2734	.3568	.4401
1/32	.0260	.1094	.1927	.2760	.3594	.4427
5/64	.0286	.1120	.1953	.2786	.3620	.4453
3/32	.0312	.1146	.1979	.2812	.3646	.4479
7/64	.0339	.1172	.2005	.2839	.3672	.4505
1/16	.0365	.1198	.2031	.2865	.3698	.4531
9/64	.0391	.1224	.2057	.2891	.3724	.4557
5/32	.0417	.1250	.2083	.2917	.3750	.4583
11/64	.0443	.1276	.2109	.2943	.3776	.4609
3/16	.0469	.1302	.2135	.2969	.3802	.4635
13/64	.0495	.1328	.2161	.2995	.3828	.4661
7/32	.0521	.1354	.2188	.3021	.3854	.4688
15/64	.0547	.1380	.2214	.3047	.3880	.4714
1/8	.0573	.1406	.2240	.3073	.3906	.4740
17/64	.0599	.1432	.2266	.3099	.3932	.4766
9/32	.0625	.1458	.2292	.3125	.3958	.4792
19/64	.0651	.1484	.2318	.3151	.3984	.4818
5/16	.0677	.1510	.2344	.3177	.4010	.4844
21/64	.0703	.1536	.2370	.3203	.4036	.4870
11/32	.0729	.1562	.2396	.3229	.4062	.4896
23/64	.0755	.1589	.2422	.3255	.4089	.4922
3/8	.0781	.1615	.2448	.3281	.4115	.4948
25/64	.0807	.1641	.2474	.3307	.4141	.4974
13/32						
27/64						
7/16						
29/64						
15/32						
31/64						
1/2						
33/64						
17/32						
35/64						
9/16						
37/64						
19/32						
39/64						
5/8						
41/64						
21/32						
43/64						
11/16						
45/64						
23/32						
47/64						
3/4						
49/64						
25/32						
51/64						
13/16						
53/64						
27/32						
55/64						
7/8						
57/64						
29/32						
59/64						
15/16						
61/64						
31/32						
63/64						
1						

DECIMALS OF A FOOT FOR EACH 1/32 OF AN INCH.

Inch.	6"	7"	8"	9"	10"	11"
0	.5000	.5833	.6667	.7500	.8333	.9167
1/32	.5026	.5859	.6693	.7526	.8359	.9193
1/16	.5052	.5885	.6719	.7552	.8385	.9219
3/64	.5078	.5911	.6745	.7578	.8411	.9245
1/8	.5104	.5937	.6771	.7604	.8437	.9271
5/64	.5130	.5964	.6797	.7630	.8464	.9297
3/32	.5156	.5990	.6823	.7656	.8490	.9323
7/64	.5182	.6016	.6849	.7682	.8516	.9349
1/4	.5208	.6042	.6875	.7708	.8542	.9375
9/32	.5234	.6068	.6901	.7734	.8568	.9401
5/16	.5260	.6094	.6927	.7760	.8594	.9427

TABLE—*Continued.*

Inch.	6"	7"	8"	9"	10"	11"
1/16	.5286	.6120	.6953	.7786	.8620	.9453
1/32	.5312	.6146	.6979	.7812	.8646	.9479
3/64	.5339	.6172	.7005	.7839	.8672	.9505
1/32	.5365	.6198	.7031	.7865	.8698	.9531
1/16	.5391	.6224	.7057	.7891	.8724	.9557
1/32	.5417	.6250	.7083	.7917	.8750	.9583
3/64	.5443	.6276	.7109	.7943	.8776	.9609
1/8	.5469	.6302	.7135	.7969	.8802	.9635
1/16	.5495	.6328	.7161	.7995	.8828	.9661
5/64	.5521	.6354	.7188	.8021	.8854	.9688
3/32	.5547	.6380	.7214	.8047	.8880	.9714
1/16	.5573	.6406	.7240	.8073	.8906	.9740
1/32	.5599	.6432	.7266	.8099	.8932	.9766
3/64	.5625	.6458	.7292	.8125	.8958	.9792
1/16	.5651	.6484	.7318	.8151	.8984	.9818
1/32	.5677	.6510	.7344	.8177	.9010	.9844
5/64	.5703	.6536	.7370	.8203	.9036	.9870
3/32	.5729	.6562	.7396	.8229	.9062	.9896
1/16	.5755	.6589	.7422	.8255	.9089	.9922
1/32	.5781	.6615	.7448	.8281	.9115	.9948
3/64	.5807	.6641	.7474	.8307	.9141	.9974
1						1.0000

THE METRIC SYSTEM.

The metric system is based on the meter, which, according to the U. S. Coast and Geodetic Survey Report of 1884, is equal to 39.370432 inches. The value commonly used is 39.37 inches, and is authorized by the U. S. Government.

There are three principal units—the meter, the liter (pronounced lee-ter) and the gram, the units of length, capacity and weight, respectively. Multiples of these units are obtained by prefixing to the names of the principal units the Greek words deka (10), hekto (100), and kilo (1000); the sub multiples, or divisions, are obtained by prefixing the Latin words deci ($\frac{1}{10}$), centi ($\frac{1}{100}$), and milli ($\frac{1}{1000}$). These prefixes form the key to the entire system. In the following tables, the abbreviations of the principal units of these sub-multiples begin with a small letter, while those of the multiples begin with a capital letter; they should always be written as here printed.

Measures of Length.

10 millimeters (mm)	=1 centimeter	cm.
10 centimeters	=1 decimeter	dm.
10 decimeters	=1 meter	m.
10 meters	=1 dekameter	Dm.
10 dekameters	=1 hektometer	Hm.
10 hektometers	=1 kilometer	Km.

Measures of Surface (not Land).

100 square millimeters (mm^2)	=1 square centimeter	cm^2.
100 square centimeters	=1 square decimeter	dm^2.
100 square decimeters	=1 square meter	m^2.

Measures of Volume.

1000 cubic millimeters (mm^3)	=1 cubic centimeter	cm^3.
1000 cubic centimeters	=1 cubic decimeter	dm^3.
1000 cubic decimeters	=1 cubic meter	m^3.

Measures of Capacity.

10 milliliters (ml)	=1 centiliter	cl.
10 centiliters	=1 deciliter	dl.
10 deciliters	=1 liter	l.
10 liters	=1 dekaliter	Dl.
10 dekaliters	=1 hektoliter	Hl.
10 hektoliters	=1 kiloliter	Kl.

NOTE.—The liter is equal to the volume occupied by 1 cubic decimeter of pure distilled water at its temperature of maximum density, or 39.2° F.

Measures of Weight.

10 milligrams (mg)	=1 centigram	cg.
10 centigrams	=1 decigram	dg.
10 decigrams	=1 gram	g.
10 grams	=1 dekagram	Dg.
10 dekagrams	=1 hektogram	Hg.
10 hektograms	=1 kilogram	Kg.
1000 kilograms	=1 ton	T.

NOTE.—The gram is the weight of 1 cubic centimeter of pure distilled water at a temperature of 39.2° F.; the kilogram is the weight of 1 liter of water; the ton is the weight of 1 cubic meter of water.

A ***formula*** is an abridged statement of a general rule, in which symbols are used.

The symbols used are the letters of the alphabet, which represent numbers, and the signs $+$, $-$, $\times$, $\div$, $\surd$, etc., which have the same meaning as in arithmetic.

To illustrate, let the following example be taken: If a person exchanges 10 books, worth \$3 per volume, for cloth at \$2 per yard, how many yards will he obtain? A rule for solving this example, and all others like it, may be stated as follows: Multiply the number of books by the price per volume, and divide the product by the price of the cloth. The result will be the number of yards of cloth.

A more concise way of stating the rule is by using letters. Thus,

Let A = the number of books.
B = the price per volume.
C = price of the cloth.
D = the number of yards of cloth.

Then, according to the rule,

$$\frac{\text{number of books} \times \text{price per volume}}{\text{price of cloth}} = \text{number of yards of cloth, or } \frac{A \times B}{C} = D.$$

This last expression is a formula; the letters A, B, C and D stand for the numbers given in the particular example to which it is applied, and the sign of the multiplication $\times$, and the horizontal dividing line of the fraction, which indicates division, show what operations must be performed upon the numbers to produce the answer D. In the example in question, A = 10, the number of books; B = 3, the price per volume; C = 2, the price of the cloth. Hence, writing for A, B and C their values, 10, 3 and 2, D, the number of yards $\frac{10 \times 3}{2} = 15$. Ans.

In modern technical works, the rules for solving examples are commonly given by formulas, and it is important to understand how to use them. Having became accustomed to them, they will often be found more convenient than rules written out in words.

The ***multiplication sign,*** $\times$, *is generally omitted in formulas, multiplication being indicated by simply writing the letters or expresions together.* Thus, the formula $\frac{A \times B}{C} = D$ given above, would ordinarily be written $\frac{A\ B}{C} = D$. The expression 4 a b means the same as $4 \times a \times b$. Evidently the sign cannot be omitted between *two figures*, as addition, instead of multiplication, would be indicated. Thus, 32 means $30 + 2$, not 3×2.

Parentheses ( ), ***brackets*** [ ], ***brace*** { } and the ***vinculum*** ———, all signify that the quantities enclosed by them

Rule for Extracting any Root of any Number.

I. Point off the number into periods which will contain as many figures as there are units in the index of the root, beginning at the decimal point.

II. Find the largest number which, when raised to the power indicated by an exponent having as many units as the index figure of the root, does not exceed the first period; the number thus obtained will be the first figure of the root.

III. Raise the first figure of the root to the power indicated by an exponent having as many units as the index figure of the root, and subtract the result from the first period; annex the first figure of the second period to the remainder, and call the result the first dividend.

IV. Raise the first figure of the root to that power indicated by an exponent which has one less unit than the index figure of the root; multiply the result by the index figure, and call the product the first divisor.

V. Divide the first dividend by the first divisor and obtain two figures of the quotient. If the second figure is 5 or a greater number, write the first figure of the quotient as the second figure of the root; if less than 5, subtract 1 from the first figure of the quotient for the second figure of the root. If the divisor is greater than the dividend, write a cipher for the second figure of the root. If the dividend contains the divisor 10 or more times, try 9 for the second figure of the root; if 9 is also too large, try 8, and so on.

VI. Raise that portion of the root already found to the power indicated by an exponent having as many units as the index figure; subtract the result from the first two periods; annex the first figure of the third period to the remainder, and call the result the second dividend.

VII. Raise that portion of the root already found to the power indicated by an exponent having one less unit than the index figure; multiply the result by the index figure, and call the product the second divisor. Divide the second dividend by

the second divisor (as described in V) for the third figure of the root.

VIII. Proceed as in VI and VII for the fourth figure of the root, and so on for more figures, if desired.

NOTE.—The result obtained in V may be too large; if so, it will be made evident in VI when getting the second dividend, and a smaller number must be used for the second figure of the root. If the given number whose root is to be found is wholly decimal, take care that the first period contains as many figures (annexing ciphers if necessary) as there are units in the index figure of the root. Thus, in extracting the 7th root of .02794, the first period would be .0279400, and the remaining periods, cipher periods.

EXAMPLE.— $\sqrt[8]{909,208,700,718,879,776} = ?$

1st period. 2d period. 3d period. 4th period.

SOLUTION.— 909 20870 07188 79776 | 3,906

1st divisor $3^8 = 243$

$= 3^4 \times 5 = 405$ | 6662 1st dividend.

16. +

Since 16 is greater than 10, we try 9.

90920870 1st and 2d periods.

$39^8 =$ 90224199

2d divisor $= 39^4 \times 5 = 11567205$ | 6961710. 2d dividend.

0

Since the divisor is greater than the dividend, we write 0 for the 3d figure of the root.

9092087007188 1st, 2d, 3d periods.

$390^8 =$ 9022419900000

3d divisor $= 390^4 \times 5 = 115672050000$ | 696171071887. 3d dividend.

6. +. Try 6.

90920870071887976. 1st, 2d, 3d and 4th periods.

$3906^8 =$ 90920870071887976

NOTE.—After having obtained the first three figures of the root, the first figure of the quotient obtained by dividing a divi-

dend by its corresponding divisor, will always be the next figure of the root.

EXAMPLE.—Extract the cube root of the above number.

SOLUTION.— (1) (2) (3) (4) (5) (6)
909 203 700 718 879 776 | 968769 +

$9^3 = 729$

1st divisor $= 9^2 \times 3 = 243$ | 1802 1st dividend.
7. +

Trying 7, it will be found to be too large. Hence try 6.

909203 1st and 2d periods.

$96^3 = 884736$

2d divisor $= 96^2 \times 3 = 27648$ | 244677 2d dividend.
8. +

909203700 1st, 2d, 3d, periods.

$968^2 = 907039232$

3d divisor. $= 968^2 \times 3 = 2811072$ | 21644687 3d dividend.
7. +

909203700718

$9687^2 = 909008405703$ 1st, 2d, 3d, 4th.

4th divisor $= 9687^2 \times 3 = 281513907$ | 1952950158 4th dividend.
6. +

909203700718879 1st,2d,3d,4th,5th

$96876^2 = 909177324509376$

5th divisor$=96876^2 \times 3=28154878128$ | 263762095037 5th dividend.
9. +

NOTE.—If more figures are desired, they may be obtained by division. The division may be commenced after the fourth figure of the root has been found by dividing the 4th dividend by the 4th divisor; the first three figures of the quotient may be written in the root. Thus, $1952950158 \div 281513907 = 693 +$, the 5th, 6th and 7th figures of the root. If still greater accuracy is desired, begin the division with the 5th dividend and 5th divisor, using the first *four* figures of the root. This note applies to any root, when the number given is not an exact power.

MENSURATION.

In the following formulas:

D = large diameter.
d = small diameter.
R = radius corresponding to D
r = radius corresponding to d
p = perimeter or circumference.
C = area of convex surface = area of flat surface which can be rolled into the shape shown.
S = area of entire surface = c plus area of the end or ends.
A = area of plane figure.
π = 3.1416.

The other letters used will be found on the engravings.

$p = \pi d = 3.1416\ d.$

$p = 2\pi r = 6.2832\ r.$

$p = 2\sqrt{\pi A} = 3.5449\sqrt{A}.$

$p = \frac{2A}{r} = \frac{4A}{d}.$

$d = \frac{p}{\pi} = \frac{p}{3.1416}.$

$d = 2\sqrt{\frac{A}{\pi}} = 1.1284\sqrt{A}.$

$r = \frac{p}{2\pi} = \frac{p}{6.2832}.$

$r = \sqrt{\frac{A}{\pi}} = 0.5642\sqrt{A}.$

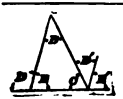
$A = \frac{\pi d^2}{4} = .7854d^2.$

$A = \pi r^2 = 3.1416r^2.$

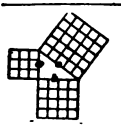
$\frac{p^2}{4\pi} = \frac{p^2}{12.5664}.$

$A = \frac{pr}{2} = \frac{pd}{4}.$

Triangles.



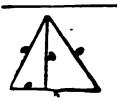
$D = B + C.$ $E + B + C = 180°.$
$B = D - C.$ $E' + B + C = 180°.$
$E' = E.$ $B' = B.$
The above letters refer to angles.



$c = \sqrt{a^2 + b^2}.$

$a = \sqrt{c^2 - b^2}.$

$b = \sqrt{c^2 - a^2}.$

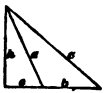


c = length of side opposite an acute angle.

$c = \sqrt{a^2 + b^2 - 2be}.$

$h = \sqrt{a^2 - e^2}.$

MENSURATION — *Continued.*



c = length of side opposite an obtuse angle.

$$c = \sqrt{a^2 + b^2 + 2be}.$$

$$h = \sqrt{a^2 - e^2}.$$

$$c : b :: a : h.$$

$$h = \frac{a\,b}{c} = \frac{c\,e}{a}.$$

$$a : b + e = e : a = h : c.$$

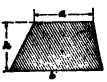
$$A = \frac{bh}{2} = \tfrac{1}{2}\,bh.$$

$$A = \frac{b}{2}\sqrt{a^2 - \left(\frac{a^2 + b^2 - c^2}{2\,b}\right)^2}.$$

Rectangle and Parallelogram.

$$A = ab.$$

$$A = b\sqrt{c^2 - b^2}.$$



Trapezoid.

$$A = \tfrac{1}{2}\,h\,(a + b).$$

Trapezium.

$$A = \tfrac{1}{2}\,[a(h' + h) + bh' + ch].$$

Ellipse.

$$p = \pi\sqrt{\frac{D^2 + d^2}{2} - \frac{(D - d)^2}{8.8}}$$

$$A = \frac{\pi}{4}\,Dd = 0.7854\,Dd.$$

MENSURATION — *Continued.*

Ring.

$$A = 0.7854\,(D^2 - d^2).$$

Sector.

$$A = \tfrac{1}{2}\,lr.$$

$$A = \frac{\pi\, r^2\, E}{360} = \frac{.008727\, r^2\, E}{l = \text{length of arc.}}$$

Segment.

$$A = \tfrac{1}{2}\,[lr - c\,(r - h)].$$

$$A = \frac{\pi\, r^2\, E}{360} \pm \frac{c}{2}\,(r - h).$$

$$l = \frac{\pi\, r\, E}{180} = 0.0175 r E.$$

$$E = \frac{180\, l}{\pi\, r} = 57.2956\,\frac{l}{r}.$$

Chord

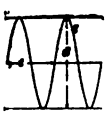
$$r = \frac{c^2 + 4h^2}{8\,h} = \frac{c^2}{2\,h}.$$

$$c = 2\,\sqrt{2hr - h^2}.$$

$$l = \frac{8\,h - c}{3} =$$

Helix.

To construct a Helix.



$$t = \sqrt{\frac{l^2}{n^2} - \pi^2 d^2}.$$

l = length of helix.
n = No. turns.
t = pitch.

$$l = n\,\sqrt{\pi^2 d^2 + t^2}.$$

$$n = \frac{l}{\sqrt{\pi^2 d^2 + t^2}}.$$

MENSURATION – *Continued.*

Cylinder.

$$C = \pi\, dh.$$
$$S = 2\,\pi\, rh + 2\,\pi\, r^2.$$
$$= \pi\, dh + \tfrac{1}{2}\,\pi\, d^2.$$
$$V = \pi\, r^2\, h = 0.7854\, d^2h.$$
$$V = \frac{p^2h}{4\,\pi} = 0.0796\, p^2h.$$

Cone.

$$C = \tfrac{1}{2}\,\pi\, dl = \pi\, rl.$$
$$S = \pi\, rl + \pi\, r^2 = \pi\, r\, \sqrt{r^2 + h^2} + \pi\, r^2.$$
$$V = \frac{0.7854\, d^2h}{3}.$$
$$= \frac{p^2h}{12\,\pi}.$$

Sphere.

$$S = 4\,\pi\, r^2 = \pi\, d^2.$$
$$S = \frac{p^2}{\pi}.$$
$$V = \frac{r}{3}\, S = 0.5236\, d^3.$$

Circular Ring.

D = mean diameter.
R = mean radius.
$$S = 4\,\pi^2\, Rr = 9.8696\, Dd.$$
$$V = 2\,\pi^2\, Rr^2 = 2.4674\, Dd^2.$$

Frustum of a Cone.

$$C = \frac{\pi\, l}{2}\,(D + d).$$
$$S = \frac{\pi\, l}{2}\,(D + d) + 0.7854\,(D^2 + d^2).$$
$$V = 1.12\,\pi\, h\,(D^2 + Dd + d^2).$$

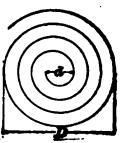
MENSURATION.—*Continued.*

Pyramid.

C = perimeter of base $\times \frac{1}{2} l$.
S = perimeter of base $\times \frac{1}{2} l$ + area of base.
V = $\frac{1}{3}$ area of base $\times h$.
To find area of base, divide it into triangles.

Frustum of a Pyramid.

$C = \frac{1}{2} l (P + p)$.
$S = \frac{1}{2} (P + p) \times l + A + a$.
$V = \frac{h}{3}\left(A + a + \sqrt{Aa} \right)$.
a = area of upper base.
A = area of lower base.
p = perimeter of upper base.
P = perimeter of lower base.



Length of Spiral.

$l = \pi n \left(\frac{D + d}{2} \right)$. n = number of coils.
l = length of spiral.

$l = \frac{\pi}{t} (R^2 - r^2)$. t = pitch.

STRENGTH OF MATERIALS.

The ultimate strengths of different materials vary greatly from the average values given in the following tables. In actual practice, the safest procedure would be to make a test of the material for its ultimate strength and coefficient of elasticity, or else specify in the contract that it shall not fall below certain prescribed limits. In the following formulas:

A = Area of cross section of material in square inches.
E = Coefficient of elasticity in pounds per square inch.
G² = Square of least radius of gyration.
I = Moment of inertia about an axis of rotation passing through the center of gravity of the section.
M = Maximum bending moment in inch pounds.
P = Total stress in pounds.
R = Moment of resistance.

S = Ultimate stress in pounds per square inch of area of section.
W = Weight placed on a beam, in pounds.
b = Breadth of cross-section of beam in inches.
d = Depth of beam in inches = diameter of circular section = altitude of triangular section = length of a vertical side.
e = Amount of elongation or shortening in inches.
f = Factor of safety.
l = Length in inches.
p = Pressure per square inch.
π = Ratio of the circumference of the circle to its diameter = 3.1416.
q = A constant used in formula for columns.
r = Radius of a circular section.
s = Elastic set or deflection in inches of a beam under a transverse (bending) stress.
t = Thickness of a shell or hollow section.

For tension, compression (where the piece does not exceed ten times its least diameter) and for shear.

$$P = \frac{AS}{f} \quad (1).$$

To find the breaking stress, P, make f = 1. For safe load, take f from Table I, and S from Table II., according to the nature and character of stress.

Table I.

FACTORS OF SAFETY, f.

Name of Material.	*Steady Stress.*	*Varying Stress.*	*Shocks (Machines).*
Cast-iron	6	15	20
Wrought Iron	4	6	10
Steel	5	7	15
Wood	8	10	15
Brick and Stone	15	25	30

Table II.

ULTIMATE STRENGTHS, S.

Name of Material.	*Tension.*	*Compression.*	*Shear.*	*Flexure.*
Cast-iron.....	20,000	90,000	20,000	36,000
Wrought Iron....	50,000	50,000	47,000	50,000
Steel.....	100,000	150,000	70,000	120,000
Wood.....	10,000	8,000	600 to 3,000	9,000
Stone.....		6,000		2,000
Brick.....	200	2,500		

For example, a square cast-iron pillar 18 inches long is required to sustain a steady load of 75,000 pounds; what must be the length of a side?

From the table, f = 6, and S = 90,000.

$$P = \frac{AS}{f} \text{ or } A = \frac{Pf}{S} = \frac{75{,}000 \times 6}{90{,}000} = 5 \text{ square inches.}$$

Length of side = $\sqrt{5} = 2.236$ inches, say $2\frac{1}{4}$ inches.

The amount of elongation or of shortening of a piece under a stress is given by the formula

$$e = \frac{Pl}{AE} \quad (2).$$

The coefficient of elasticity, E, must be taken from the following table:

Table III.

Name of Material.	*Coefficient of Elasticity.*	*Elastic Limit for Tension.*
Cast-iron.....	15,000,000	6,000
Wrought Iron.....	25,000,000	25.000
Steel.....	30,000,000	50,000
Wood.....	1,500,000	3,000

A wrought iron bar 24 feet long, 1½ inches in diameter would elongate under a tensile stress of 15 tons

$$e = \frac{(15 \times 2{,}000) \times (24 \times 12)}{\frac{1}{4}\pi(1\frac{1}{2})^2 \times 25{,}000{,}000.} = .196 \text{ inch.}$$

To find the breaking strength of a beam, use the formula

$$M = SR \quad (3).$$

Obtain M and R from the two following tables, according to the kind of beam and nature of cross-section. A simple beam is one merely supported at its ends. In the expressions for R, d is always understood to be the vertical side or depth; hence, that beam is the stronger which always has its greatest depth or longest side vertical.

Table IV.

Kind of Beam and Manner of Loading.	*Bending Moment.* M	*Deflection.* s
Cantilever, load at end.....	Wl	$\frac{1}{3}\frac{Wl^3}{EI}$
Cantilever, uniformly loaded..	½ Wl	$\frac{1}{8}\frac{Wl^3}{EI}$
Simple Beam, load at middle..	¼ Wl	$2\frac{1}{6}\frac{Wl^3}{EI}$
Simple Beam, uniformly loaded	⅛ Wl	$2\frac{4}{6}\frac{Wl^3}{EI}$
Beam fixed at both ends, load at middle.....	⅛ Wl	$1\frac{1}{6}\frac{Wl^3}{EI}$
Beam, fixed at both ends, uniformly loaded.....	$\frac{1}{12}$ Wl	$2\frac{4}{6}\frac{Wl^3}{EI}$

Table V.

ne of Section	I	R	G²
cular	$\frac{\pi d^4}{64}$	$\frac{\pi d^3}{32}$	$\frac{d^2}{16}$
circu-	$\frac{\pi (d^4-d_1{}^4)}{64}$	$\frac{\pi (d^4-d_1{}^4)}{32d}$	$\frac{d^2+d_1{}^2}{16}$
ıare	$\frac{d^4}{12}$	$\frac{d^3}{6}$	$\frac{d^2}{12}$
quare	$\frac{d^4-d_1{}^4}{12}$	$\frac{d^4-d_1{}^4}{6d}$	$\frac{d^2+d_1{}^2}{12}$
ctan-	$\frac{bd^3}{12}$	$\frac{bd^2}{6}$	$\frac{b^2}{12}$
rect- r	$\frac{bd^3-b_1d_1{}^3}{12}$	$\frac{bd^3-b_1d_1{}^3}{6d}$	$\frac{b^2d-b_1{}^2d_1}{12(bd-b_1d_1)}$
angu-	$\frac{bd^3}{36}$	$\frac{bd^2}{24}$	$\frac{d^2}{18}$
lliptl-	$\frac{\pi bd^3}{64}$	$\frac{\pi bd^2}{32}$	$\frac{b^2}{64}$
ellip-	$\frac{\pi}{64}-(bd-b^3{}_1d_1{}^3)$	$\frac{\pi (bd^3-b_1d_1{}^3)}{32d}$	$\frac{b^3d-b_1{}^3d_1}{64(bd-b_1d_1)}$
a	$\frac{bd^3-b_1d_1{}^3}{12}$	$\frac{bd^3-b_1d_1{}^3}{6d}$	$\frac{b^3d\ b_1{}^3d_1\ )}{12(bd-b_1d_1)}$
vith arms oxi-			$\frac{.1d^2}{24}$
vith arms oxi-			$\frac{.1d^2}{24}$

Thus the breaking strength of a cast iron simple beam uniformly loaded and 20 feet long between the supports, having a hollow rectangular cross-section 8 inches by 6 inches outside and 6 inches by 4 inches inside, is,

$$M = SR \text{ or } \tfrac{1}{8}\,Wl = 36000 + \frac{bd^3 - b_1 d_1^3}{6d}$$

$$\text{or } W = \frac{36000 \times 8 \times (6 \times 8^3 - 4 \times 6^3)}{(20 \times 12) \times (6 \times 8)} = 55200.$$

Using a factor of safety of 6, the beam should support

$$\frac{55200}{6} = 9200 \text{ pounds}$$

with perfect safety. The value S for beams should be taken from the flexure column of Table II.

To find the amount of deflection in a beam due to a load, substitute the values of W, l, E and I in the different expressions for the deflection s in Table IV.

The value of I is to be taken from Table V.

For example, what is the deflection of a wrought iron beam, fixed at both ends, 7 feet long between the supports, having a solid rectangular cross-section 6 inches wide and 2¾ inches deep, carrying a load of 21,000 pounds in the middle?

From the table

$$s = \frac{Wl^3}{192\,EI} = \frac{Wl^3}{192E \times \frac{bd^3}{12}} = \frac{21000 \times (7 \times 12)^3 \times 12}{192 \times 25{,}000{,}000 \times 6 \times (2\frac{3}{4})^3} = .249''.$$

Columns.

To find the breaking strength of a column use the following formula:

$$P = \frac{SA}{1 \times q \frac{l^2}{G^2}} \qquad (4).$$

S is taken from Table II, in the column for compression, G^2 from Table V, and q from the following table, according to the kind and condition of the ends.

sufficient metal must be added at that point to provide means of attaching the weight.

Shafting.

The diameter of a shaft may be found by the following formula. The first is used when great stiffness is required, and the shafts very long; the second when strength only is required to be considered.

d = Diameter of shaft in inches.
H = Horsepower transmitted.
N = Number of revolutions per minute.
c = Constant in formula 5.
k = Constant in formula 6.

$$d = c\sqrt[4]{\frac{H}{N}} \quad (5). \qquad d = k\sqrt[4]{\frac{H}{N}} \quad (6).$$

c = 5.26 for cast iron, 4.75 for wrought iron and 3.96 for steel.
k = 4.02 for cast iron, 3.63 for wrought iron and 3.03 for steel.

NOTE.—*To extract the fourth root, extract the square root twice.*

Pipes and Cylinders.

p = Pressure in pounds per square inch.
d = Diameter of pipe or cylinder in inches.
t = Thickness in inches.
S = Ultimate tensile strength taken from Table II.
r = Inside radius in inches.
f = Factor of safety, usually taken as 6 for wrought iron and 12 for cast iron.

For thin pipes $p.d\ f = 2\ t\ S$ (7).

For thick pipes or cylinders.

$$p\ f = \frac{S\ t}{r + t} \quad (8).$$

Ropes and Chains.

D = Diameter in inches of the rope = diameter of iron from which the link in chain is made.

W = Safe load in tons of 2,000 pounds.

For common hemp rope, $W = \frac{1}{3} D^2$.
For iron wire rope, $W = \frac{1}{8} D^2$.
For steel wire rope, $W = \frac{1}{15} D^2$.
For close link wrought iron chain, $W = 6 D^2$.
For stud link wrought iron, $W = 9 D^2$.

MECHANICS.

Falling Bodies.

Let g = 32.16 = Constant accelerating force due to the attraction of the earth.
t = Number of seconds that the body falls.
V = Velocity in feet per second at the end of the time, t.
h = Distance that a body falls during the time, t.

$$\text{Then, } V = gt = \frac{2h}{t} = \sqrt{2gh} = 8.02\sqrt{h.}$$

$$h = \frac{V\,t}{2} = \frac{gt^2}{2} = \frac{V^2}{2g} = .015547\,V^2$$

$$t = \frac{V}{g} = \frac{2\,h}{V} = \sqrt{\frac{2\,h}{g}} = .24938\sqrt{h.}$$

Projectiles.

Let g = 32.16 = Acceleration due to gravity.
v = Initial velocity in feet per second.
r = Range.
y = Vertical height of starting point above ground.
A = Elevation in degrees = the angle which the direction of the projectile at the start makes with the horizontal.

Then $r = \frac{v^2}{g} \sin 2A$ = the range on a horizontal line through the starting point.

If the body is projected in a horizontal direction,

$$r = v\sqrt{\frac{2\,y}{g}} = .24938\ v\ \sqrt{y.}$$

The range of a projectile fired in a horizontal direction, 80 feet above the ground, with a velocity of 300 feet per second is

$r = .24938 \times 300 \times \sqrt{80} = 409.77$ feet.

Centrifugal Force.

F = Centrifugal force in pounds.

W = Weight of revolving body in pounds.

r = Distance from the axis of motion to the center of gravity of the body in feet; for a flywheel take it equal to the inside radius of the rim.

N = Number of revolutions per minute.

V = Velocity in feet per second.

$$F = \frac{W\ V^2}{gr} = .00034\ Wr\ N^2.$$

The amount of centrifugal force in a flywheel weighing 7 tons, making 150 revolutions per minute and having an inside radius of 5 feet, is $F = .00034 \times (7 \times 2000) \times 5 \times 150^2 = 535.500$ pounds.

Center of Gravity.

The center of gravity of a body, or of a system of bodies, is that point from which, if the body or system were suspended, it would be in equilibrium.

If a line or a surface has two axes, or a solid has three axes of symmetry, the center of gravity lies at their point of intersection, and corresponds with the geometrical center of the figure.

An axis of symmetry is any line so drawn, that if part of the figure on one side of the line be folded on this line, it will coincide exactly with the other part, point for point and line for line. Thus in Fig. 1, if the part *a b* be folded on the line

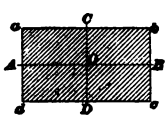


FIG. 1.

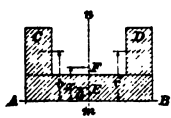


FIG. 2.

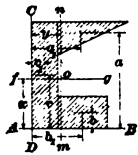


FIG. 3.

A B, the upper half will coincide exactly with the lower half: also, if *b c* be folded on the line *C D*, the right-hand half will coincide exactly with the left-hand half. Hence, the point *O* where *A B* and *C D* intersect is the center of gravity of the

rectangle $a\ b\ c\ d$. If the figure has one axis of symmetry, the center of gravity may be found as follows: Let $m\ n$ be an axis of symmetry of the area in Fig. 2. The center of gravity will lie somewhere on this line. Draw any line, $A\ B$, perpendicular to $m\ n$. Divide the area into squares, rectangles, triangles, parallelograms, circles, etc., whose centers of gravity are easily found, and measure the perpendicular distances of these centers of gravity from the line $A\ B$. Add the sum of the areas, multiplied by the distance of their centers of gravity from the line $A\ B$ and divide by the area of the entire figure; the result is the distance x of the center of gravity from $A\ B$ measured on $m\ n$, or the point F.

If the figure has no axis of symmetry, as in Fig. 3, draw any line, as $A\ B$, and find the distance x of the center of gravity from $A\ B$, and through x draw $f\ g$ parallel to $A\ B$. Choose any other line, $C\ D$, and find the distance y of the center of gravity from $C\ D$ by the same method, and through y draw $m\ n$ parallel to $C\ D$. The point of intersection o of $f\ g$ and $m\ n$ is the center of gravity.

Thus, suppose that the area of the triangle is A sq. in., and the distance of its center of gravity from $A\ B$ is a in., and from $C\ D$ a_1 in.; that the area of the small rectangle is B sq. in., and the distance of its center of gravity from $A\ B$ is b in., and from $C\ D$ is b_1 in.; that the area of the large rectangle is C sq. in., and the distance of its center of gravity from $A\ B$ is c in., and from $C\ D$ is c_1 in., then

$$x = \frac{(A \times a) + (B \times b) + (C \times c)}{A + B + C}, \text{ and}$$

$$y = \frac{(A \times a_1) + (B \times b_1) + (C \times c_1)}{A + B + C}.$$

To find the center of gravity mechanically, suspend the object from a point near its edge and mark on it the direction of a plumb line from that point; then suspend it from another point and again mark the direction of a plumb line. The intersection of these two lines will be directly over the center of gravity.

The center of gravity of a body having parallel sides may be found by drawing the outline of one of the sides upon heavy paper, and cutting out the exact shape of the figure. Then

suspend the paper from the two points and find the center of gravity as in the last case.

The center of gravity of a triangle lies on a line drawn from a vertex to the middle point of the opposite side, and at a distance from that side equal to $\frac{1}{3}$ of the length of the line. Or, draw a line from another vertex to the middle point of the side opposite, and the intersection of the two lines will be the center of gravity.

For a parallelogram, the center of gravity is at the intersection of the two diagonals.

For an irregular four-sided figure, draw the two diagonals; find the centers of gravity of each of the four triangles. Join the centers of gravity of each opposite pair of triangles by straight lines, and they will intersect in the center of gravity of the figure.

For a greater number of sides, find the center of gravity by one of the preceding methods.

For an arc of a circle, the center of gravity lies on the radius perpendicular to the arc (an axis of symmetry) and at a distance from the center equal to the length of the chord multiplied by the radius and divided by the length of the arc.

For a semicircle, the distance from the center = .6367 r. r = the radius.

For the area included in a half-circle, the distance of the center of gravity from the center equals .424 r.

For a circular sector, the distance of the center of gravity from the center equals two-thirds of the length of the chord times the radius, divided by the length of the arc.

For a circular segment, let A be its area and C the length of its chord; then the distance of the center of gravity from the center of the circle is equal to $\frac{C^3}{12\,A}$.

For a solid having three axes of symmetry, all perpendicular to each other, like a sphere, cube, right parallelopiped, etc., their point of intersection is the center of gravity.

For a cone or pyramid, draw a line from the apex to the middle point of the base; the center of gravity is one-fourth the length of this line from the base, measured on the line.

For two bodies, the larger weighing W lbs. and the smaller

P lbs., respectively, the center of gravity will lie on the line joining the centers of gravity of the two bodies and at a distance from the larger body equal to $\frac{P\,a}{P+W}$ where a is the distance between the two centers of gravity of the bodies.

For any number of bodies, first find the center of gravity of two of them as above, and consider them as one weight whose center of gravity is at the point just found. Find the center of gravity of this combined weight and a third body. So continue for the rest of the bodies, and the last center of gravity will be the center of gravity of all of the bodies.

The Moment of Inertia.

The *moment of inertia* of a body or section is a mathematical expression which is much used in computations relating to rotating bodies.

It may be defined as follows :

The moment of inertia of a body rotating about a given axis is the sum of the products of the weights of the elementary particles of which it is composed, by the square of their distances from the axis.

It is often desirable to use the moment of inertia for a plane section ; but as a plane suface has no weight it is apparent that the above definition does not correctly apply. The following definition applies to plane surfaces.

The moment of inertia of a plane surface about a given axis is the sum of the products of the elementary areas into which it may be conceived to be divided, by the squares of their distances from the axis.

The axis about which the body or surface rotates, or is assumed to rotate, *i. e.*, the axis from which the distance to each area or particle is measured, is called the *axis of rotation.* The least moment of inertia is that value of the moment of inertia of a body or section when the axis of rotation passes through the center of gravity, since its value is less for that position of the axis than for any other.

To find the least moment of inertia of a body rotating about ***a given axis :***

Divide the body or section into many small parts and multiply the weight or area of each part by the square of its distance from its center of gravity to the axis of rotation; the sum of these products will be the moment of inertia.

NOTE.—The results obtained by the above rules are really only approximate; for practically it is impossible to divide a body or surface into parts sufficiently small for absolute accuracy. The smaller the parts the more accurate will be the result; but the results obtained by these rules will always be *slightly too small.*

The moment of inertia is usually designated by the letter I.

Formulas for the values of I about an axis of rotation passing through the center of gravity of the section are given for various forms of sections in Table V, page 45.

The moment of inertia about an axis of rotation not passing through the center of gravity is *equal to the moment of inertia about a parallel axis through the center of gravity plus the product of the entire weight of the body (or area of the section) multiplied by the square of the distance between the two axes.*

EXAMPLE.—It is desired to find the moment of inertia of a 6 in. I beam of the dimensions shown in Fig. 1 about an axis x y perpendicular to the web of the beam at the center.

Since the axis about which the moment of inertia is to be found is an *axis of symmetry* of the beam, it is necessary to make the computations only for the half section of the beam lying at one side of the axis, and multiply the result by 2. As stated above, the smaller the parts into which the area is divided the more accurate will be the result.

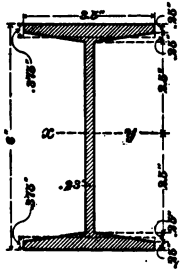


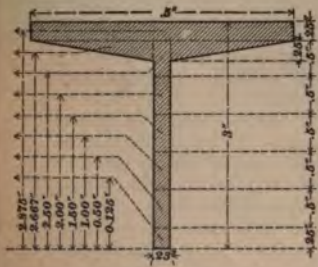
Fig. 1.

It will be sufficiently accurate for present purposes to divide the section in the manner shown in Fig. 2.

The operations are given at the side of the figure and will be

readily understood. The sum of the product is the approximate value of the moment of inertia of this half of the section about the axis x y, and when multiplied by 2 is the approximate value of I for the entire section.

It is found to equal 23.444 in.



	Areas.	Square of Distance.
3.50× 0.25=0.875	0.875×	2.875^2=7.
3.27×0.125=0.409	0.409×	2.667^2=2.
0.23× 0.50=0.115	0.115×	2.50^2=0.
0.23× 0.50=0.115	0.115×	2.00^2=0.
0.23× 0.50=0.115	0.115×	1.50^2=0.
0.23× 0.50=0.115	0.115×	1.00^2=0.
0.23× 0.50=0.115	0.115×	0.50^2=0.
0.23× 0.25=0.058	0.058×	0.125^2=0.
	1.917	11.
	2	
	A = 3.834	I = 23.

Fig. 2.

If the web of the beam be divided into areas ¼ inch in height (instead of ⅛ inch), the value of I obtained will be 23.46 inches. If the section is considered to be of the form indicated by the dotted lines in Fig. 1, and having the same area as the original section then, by the formula for the moment of inertia of an I beam given in Table V, page 45, the value of

$$I = \frac{3.50 \times 6^3 - 3.27 \times 5.25^3}{12} = 23.57 \text{ in.}$$

The true value is almost exactly 23.48 inches. Any one of these values would be sufficiently correct for most practical purposes.

If it is desired to find the moment of inertia of a body about a given axis with reference to the *weight* of the body, the process is substantially the same as in the example given for the plane section, except that the *weight* of each small part of the

body is taken instead of the *area* of each small part of the section.

Center of Oscillation.

The *center of oscillation* of a pendulum or other body vibrating or rotating about a fixed axis or center, is that point at which, if the entire weight of the body were concentrated, the body would continue to vibrate in the same intervals of time.

When a pendulum, or other suspended body, is oscillating backward and forward, it is plain that those particles which are farther from the point of suspension travel through greater distances, and therefore move with greater velocities than those particles which are nearer the point of suspension.

But there is evidently some point on the pendulum which travels through the same distance and has the same velocity as the average distance and average velocity of all the particles. This point is called the *center of oscillation;* it is *not* situated at the center of gravity.

A center of oscillation always exists in the ball of a revolving governor or other rotating body.

The axis or center around which the body rotates (corresponding to the point of suspension in the pendulum) is the *axis of rotation.*

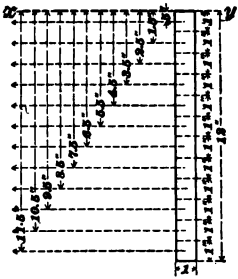
The distance from the axis, or center of rotation, to the center of oscillation is sometimes called the *true length of the pendulum;* it is also called the *radius of oscillation;* the latter name is preferable.

To find the radius of oscillation: *Divide the moment of inertia of the body about the given axis of rotation by the product of the total weight of the body, multiplied by the distance from the given axis to the center of the gravity of the body.*

The center of oscillation and the center of rotation (point of suspension) are *interchangeable.* If the position of a pendulum is reversed and suspended from the center of oscillation, it will vibrate in the same intervals of time.

EXAMPLE.—It is desired to find the position of the center of

oscillation of a wrought iron bar 1 in. sq. and 12 in. long, axis of rotation perpendicular to the bar at one end :



Weight of each cubic inch.		*Square of Distance.*	
0.281	×	0.5^2	= 0.070
0.281	×	1.5^2	= 0.632
0.281	×	2.5^2	= 1.756
0.281	×	3.5^2	= 3.442
0.281	×	4.5^2	= 5.690
0.281	×	5.5^2	= 8.500
0.281	×	6.5^2	= 11.872
0.281	×	7.5^2	= 15.806
0.281	×	8.5^2	= 20.302
0.221	×	9.5^2	= 25.360
0.281	×	10.5^2	= 30.980
0.281	×	11.5^2	= 37.162
3.372			161.572 = I

For the purposes of the example it will be sufficiently accurate to find the moment of inertia by considering the bar to be divided into 12 equal cubes, each containing 1 cu. in. of metal, as indicated in the figure, and the weight of each cube to be concentrated at its center of gravity.

The weight of a cu. in. of wrought iron is .281 lb., and of a bar 1 in. sq. and 1 ft. long it is $.281 \times 12 = 3.372$ lbs. Hence, $I = .281 \times 0.5^2 + .281 \times 1.5^2 +$ etc. = 161.572. (See figure.) The exact value of I is 161.856 ; this shows that the approximate method is very close.

According to the rule given above, if the moment of inertia is divided by the product of the weight of the body, by the distance from the axis of rotation to the center of gravity, the quotient will be the radius of oscillation.

Therefore, the distance from the exact center of oscillation of

a wrought iron bar, 1 in. sq. and 12 in. long, to an axis of rotation perpendicular to the end of the bar, is

$$\frac{161.856}{3.372 \times 6} = 8 \text{ in.},$$

or ⅔ the length of the bar.

The value of I for a bar of any cross-section, provided it is uniform throughout its length, revolving about an axis perpendicular to it and passing through its end, is $\frac{W\ l^2}{3}$, in which W = the weight of the bar in pounds and l is its length in inches.

$$\text{Hence, } I = \frac{W\ l^2}{3} = \frac{3.372 \times 12^2}{3} = 161.856.$$

If the axis passes through the center of gravity of the bar, $I = \frac{W\ l^2}{12}$.

Center of Percussion.

The *center of percussion* with respect to a given axis of rotation may be defined as the point of application of the resultant of the forces which cause the body to rotate. It is that point at which if a force is applied it will have no effect at the axis of rotation.

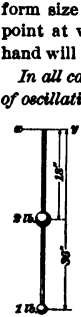
Strike anything solid, as an anvil, with a stick. If the end of the stick hits the anvil, the opposite end will sting your hand and will jerk in the direction in which the blow is struck; if the center of the stick hits the anvil it will again sting your hand, but it will jerk it in a direction opposite to the movement of the blow. But somewhere between the end and the center of the stick will be a point where it may hit the anvil and not sting your hand at all. This point is the center of percussion.

Level off the surface of some wet sand and lay a strip of board upon it (say 18 inches long and 3 inches wide). Strike or press the board near the center and the entire length of the board will be imprinted in the sand; but press it near one end and the opposite end will be raised up from the sand and will make no imprint. Between the center and end of the board is a point which if pressed upon will cause no movement in the

opposite end, *i. e.*, the end of the board will neither press into the sand nor be lifted from it, but the imprint in the sand will diminish to zero at the end of the board. The point pressed or struck will be the center of percussion. If the board is of uniform width, the center of percussion will be at ⅔ of the distance from one end of the board.

Similarly in the preceding illustration, if the stick is of uniform size and weight and your hand grasps it at one end, the point at which it can strike the anvil without affecting your hand will be at ⅔ the distance from the opposite end.

In all cases the center of percussion is identical with the center of oscillation, and its position is found in the same manner.



EXAMPLE.—It is desired to find the position of the center of oscillation or percussion of two balls fastened upon a rod. The first weighing 2 lb., is at a distance of 18 in. from the axis of rotation, and the second weighing 1 lb., is at a distance of 36 in. from the axis. See figure.

For simplicity the rod will be assumed to have no weight. Consider the weight of each ball to be concentrated at its center of gravity.

The moment of inertia is found as follows:

Wgt.		*Sq. of Dist.*		
2	×	18^2	=	648
1	×	36^2	=	1296
				1944 = I.

The center of gravity of the two balls is found to be at a distance of 6 in. from the larger, or 24 in. from the axis of rotation (see page 54), and the combined weight of the two balls is $2 + 1 = 3$ lbs. Therefore the center of percussion is found to be at a distance of $\frac{1944}{3 \times 24} = 27$ in. from the axis of rotation.

But, in an actual case, the rod would have weight, and its *moment of inertia must* be considered as well as the moment of *inertia of the balls.*

If we assumed that the rod is of steel, $\frac{3}{8}$ in. in diameter and 36 in. long, it will weigh $\left(\frac{3}{8}\right)^2 \times .7854 \times 36 \times .283 = 1.125$ lbs. .283 lb. is the weight of one cubic inch of steel.

Using the formula given on page 59:

$$I = \frac{Wl^2}{3} = \frac{1.125 \times 36^2}{3} = 486.$$

Adding this result to the former, $1944 + 486 = 2430$ = moment of inertia of rods and balls. The center of gravity of the combination is found by the formula (see page 54), $\frac{Pa}{P+W}$. Substituting, $\frac{1.125 \times 6}{1.125+3} = 1\frac{7}{11}$. $24 - 1\frac{7}{11} = 22\frac{4}{11}$ in. = distance from end of rod to center of gravity.

Applying the rule given for finding the center of oscillation, the distance of the center of percussion from the end of the bar is $\frac{2430}{(1+2+1.125) \times 22\frac{4}{11}} = 26.34$ in. very nearly.

Center of Gyration—Radius of Gyration.

The *center of gyration* is that point in a revolving body at which, if the entire mass of the body were concentrated, the moment of inertia with respect to a given axis would be the same as in the body.

An ounce of cork occupies about 94 times as much space as an ounce of platinum; but the ounce of platinum could have the same moment of inertia as the ounce of cork, if its center of gyration has the same position with respect to the axis of rotation.

The center of gyration is not at the center of gravity, nor at the center of oscillation, but at some point in a straight line between those centers.

The *radius of gyration* is the distance from the axis of rotation to the center of gyration.

The square of the radius of gyration is the average of the squares of the distance from the axis of rotation to each elementary particle of the body, or to each elementary area of the section, as the case may be. But the sum of these squares of

distances multiplied by the weight or area of each elementary part equals the moment of inertia; therefore, the moment of inertia divided by the weight of the body or area of the section equals the square of the radius of gyration; the square root of this quotient is the radius of gyration.

But according to the rule for finding the radius of oscillation, the quotient obtained by dividing the moment of inertia by the weight or area equals the product of the distance from the axis of rotation to the center of gravity multiplied by the radius of oscillation; and therefore *the radius of gyration is a mean proportional between these distances.*

If the distance from the axis of rotation to the center of gravity is known, and the radius of oscillation is known, the radius of gyration may be found by multiplying these two known distances together and extracting the square root of the product.

In the example of the I beam, Fig. 2, page 56, the sum of the areas of the half section of the beam is 1.917, and the area of the entire section is 3.834 square inches. Therefore the radius of gyration of this beam about an axis through the center of gravity perpendicular to the web $= \sqrt{\frac{23.44}{3.834}} = 2.47$ inches.

In the example of the iron bar 12 in. long, see figure, page 58, the distance from the axis of rotation to the center of gravity is 6 in., and the radius of oscillation was found to equal 8 in. Therefore the radius of gyration about an axis perpendicular to the bar at one end $= \sqrt{6 \times 8} = 6.93$ in. Or, the moment of inertia of the bar $= 161.586$ inch-pounds, and the weight of the bar $= 3.372$ pounds. Therefore, the radius of gyration $= \sqrt{\frac{161.586}{3.372}} = 6.93$ in., very nearly.

The radius of gyration is used in determining the strength of columns. The axis must be taken in such direction that the result will be the *least* radius of gyration of the column; this *condition is usually* obtained when the axis is perpendicular to *the least diameter* or side of the column.

The various relations between these quantities may be concisely expressed by the following formulas, in which

A = area of the section (or weight of the body if the weight is used).

g = distance from the axis of rotation to the center of gravity.

G = Radius of gyration.

r_o = Radius of oscillation.

I = Moment of inertia.

Then :

$I = A G^2$.	$I = A g r_o$.	$G^2 = g r_o$.
$G = \sqrt{\frac{I}{A}}$.	$g = \frac{I}{A r_o}$.	$r_o = \frac{I}{A g}$
$G = \sqrt{g r_o}$.	$g = \frac{G^2}{r_o}$	$r_o = \frac{G^2}{g}$.

$g : G :: G : r_o$.

To find the Radius of Oscillation, Radius of Gyration and Moment of Inertia Experimentally :

The connecting rod of an engine is represented in the figure. It is desired to find the moment of inertia of the rod about an axis of rotation through the center of the cross-head pin A.

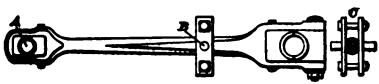


FIG. 9.

This may be accomplished experimentally as follows :

Suspend the rod from the cross-head pin in such manner that it will swing freely ; cause it to swing or oscillate, and note the exact time of the vibrations. Remove the cross-head pin and reverse the rod, but instead of suspending it by the crank pin, suspend it by a movable pin B, that can be clamped at any *desired point upon the rod.* C is another view of this pin. The

ll be a point on the rod from which it may be suspended by eans of the movable pin, so that it will vibrate in exactly the ime intervals of time as when suspended from the cross-head in. This point is the *center of oscillation*, for the center of scillation and the center of rotation are interchangeable; the point will be found at about ⅔ the length of the rod from the crank pin. Find this center of oscillation experimentally, and carefully measure the distance from the center of the movable pin to the center of the cross-head pin hole. This distance is the *radius of oscillation* = r_o. Next remove the moveable pin and find the center of gravity (lengthwise) of the rod by balancing it across a knife-edge, and measure the distance from the center of gravity thus found to the center of the cross-head pin hole; this distance = g. Finally weigh the rod.

The product of the weight (= A), the radius of oscillation (= r_o) and the distance from the center of cross-head pin (axis of rotation) to the center of gravity (= g) will be the moment of inertia. For by the formula, $I = A\,g\,r_o$. The radius of gyration, G, may be found by the formula $G = \sqrt{\frac{I}{A}}$ or by $G = \sqrt{g r_o}$.

Moment of Resistance.

If the moment of inertia of the cross-section of a beam divided by the distance from the neutral axis to the extre fiber, *i. e.*, the fiber that is farthest from the axis, the quoti will be the quantity known as the *moment of resistance.*

It is evident, that if a beam is strained by a vertical load greatest strain will be in the extreme upper and lower fibe the beam.

The intensity of the strain that can be borne by the ex fibers is the limit of the strength of the beam.

The upper fibers are compressed and the lower fibe *stretched, but somewhere* along or near the center of a

section of the beam the fibers are neither extended nor compressed; the position of these fibers is called the *neutral surface*, and the line where this neutral surface intersects a right section of the beam is the *neutral* axis of the section.

The neutral axis is usually considered to pass through the center of gravity of the section.

If the moment of resistance is multiplied by the amount of stress which may be allowed per sq. in. upon the extreme fiber, the product will represent the efficiency of the beam to resist bending moment.

EXAMPLE.—Referring to the 6-inch I beam, Figs. 1 and 2, pages 55 and 56, for which the moment of inertia of the section has been found: It is desired to ascertain the load which a wrought iron beam of the same dimensions as Fig. 1 will carry at the center of a span 8 ft. between supports.

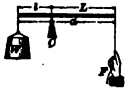
The moment of resistance for the section $= \frac{23.48}{3} = 7.83$ in.

In Table II, page 43, the ultimate strength or fiber stress for wrought iron is given at 50,000 lbs. per sq. in., and in Table I, page 42, the factor of safety given for wrought iron under a steady stress is 4; therefore the safe fiber stress for wrought iron $= \frac{S}{f} = 50,000 =$ 12,500 lbs. per sq. in., and the moment of resistance multiplied by the safe fiber stress, or $\frac{S\ R}{f}$, $= 7.83 \times 12,500 = 97,875$ inch-pounds. But l = 8 ft., or 96 ins., and equating the bending moment for a load at the center of a beam $\left(= \frac{W\ l}{4}\right)$ with the moment of resistance, or, putting $M = \frac{S\ R}{4} = \frac{W\ l}{4}$.

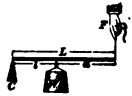
$$\frac{96\ W}{4} = 97,875$$

W = 4,078 lbs., or the load that can be safely supported at the center of the beam.

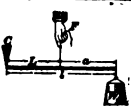
MECHANICAL POWERS.



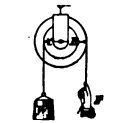
$$F : W :: l : L. \quad FL = Wl.$$
$$F = \frac{Wl}{L}. \quad W = \frac{Fl}{l}.$$
$$l = \frac{Fa}{W+F}. \quad L = \frac{Wa}{W+F}.$$



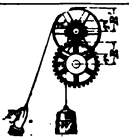
$$F : W :: l : L. \quad FL = Wl.$$
$$F = \frac{Wl}{L}. \quad W = \frac{Fl}{l}.$$
$$L = \frac{Wa}{W-F}. \quad l = \frac{Fa}{W-F}.$$



$$F : W :: l : L. \quad FL = Wl.$$
$$F = \frac{Wl}{L}. \quad W = \frac{FL}{l}.$$
$$L = \frac{Wa}{F-W}. \quad l = \frac{Fa}{F-W}.$$



$$F : W :: r : R. \quad FR = Wr.$$
$$F = \frac{Wr}{R}. \quad R = \frac{Wr}{F}.$$
$$W = \frac{RF}{r}. \quad r = \frac{RF}{W}.$$



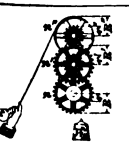
$$F = \frac{W r r'}{R R'}. \quad W = \frac{F R R'}{r r'}.$$

n = number of revolutions of large gear.

$n : n' :: r' : R$. $v : v' :: r r' : R R'$.

v = *velocity of* W.

v' = *velocity of* F.



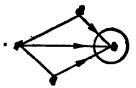
$$F = \frac{W r r' r''}{R R' R''}. \quad W = \frac{F R R' R''}{r r' r''}.$$

$n : n' :: r' r'' : R R'$.

$v : v' :: r r' r'' : R R' R''$.

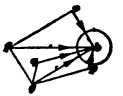
$r r' r''$, etc., = radii of the pinions.

$R R' R''$, etc., = radii of the wheels.

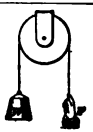


MECHANICAL POWERS—*Continued.*

Let $d\ b$ and $q\ b$ represent the magnitudes and directions of two forces which act to move the body b. By completing the parallelogram there will be obtained a diagonal force $f\ b$, whose magnitude and direction are equal to the effect produced by $d\ b$ and $q\ b$. $f\ b$ is called the resultant of $d\ b$ and $q\ b$.



If three or more forces act in different directions to move a body b, find the resultant of any two of them, and consider it as a single force. Between this and the next force find a second resultant. Thus: $p\ b$, $q\ b$ and $r\ b$ are magnitudes and directions of the forces. $p\ b + q\ b = g\ b + r\ b = f\ b$, the magnitude and direction of the three forces, $p\ b$, $q\ b$ and r b.

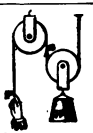


A single fixed Pulley.

$F = W.$

$v = v'.$

v = *velocity of* W; v' = *velocity of* F.



A single movable Pulley.

$F : W :: 1 : 2$, *or* $F = \frac{1}{2}\ W.$

If the force F be applied at a and act upwards, the result will be the same.

$v' = 2\ v.$

v = *velocity of* W; v' = *velocity of* F.

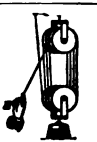
A double movable Pulley.

$F : W :: 1 : 4$, *or* $F = \frac{1}{4}\ W.$

Let u = number of parts of rope not counting the free end.

$F = w + u.$ $v : v' :: 1 : u.$

v = *velocity of* W; v' = *velocity of* F.



Quadruple movable Pulley.

$F = \frac{1}{8}\ W.$ $F : W :: 1 : 8.$

Let u = number of parts of rope, not counting the free end; then, $F = W + u.$ $v : v' :: 1 : 2\ u.$

v = *velocity of* W; v' = *velocity of* F.

MECHANICAL POWERS—*Continued.*

Compound Pulley.

u = number of movable pulleys.

$$F = \frac{W}{2^u}. \quad W = 2^u F.$$

$$v : v' :: 1 : 2^u.$$

v = *velocity of* W; v' = *velocity of* F.

Differential Pulley.

$$W = \frac{2PR}{R - r}.$$

An oblique fixed Pulley.

$$F : W :: 1 : 2 \cos z$$

$$W = 2F \cos z. \quad F = \frac{2 \cos z}{W}$$

Inclined Plane.

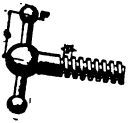
$$F = \frac{Wh}{l} = W \sin v.$$

$$W = \frac{Fl}{h} = \frac{F}{\sin v}.$$

Wedge.

$$F = \frac{R\,a}{l}. \quad R = \frac{Fl}{a}.$$

F = force required to drive the wedge.
R = resistance.



Screw.

P = pitch of the screw.
r = radius on which the force F acts.

$$F : W :: P : 2\pi r.$$

$$F = \frac{WP}{2\pi r}. \quad W = \frac{F 2\pi r}{P}.$$

Work.

Work is the overcoming of resistance through a distance. The unit of work is the *foot-pound*, that is, it equals one pound raised vertically one foot. The amount of work done is equal to the resistance in pounds, multiplied by the distance in feet through which it is overcome. If a body is lifted, the resistance is the weight or the overcoming of the attraction of gravity, the work done being the weight in pounds multiplied by the height of the lift in feet. If a body moves in a horizontal direction, the work done is the friction overcome, or the force needed to move a resistant body or combination of bodies, times the distance moved. In order to compare the different amounts of work done by different systems of forces, time is also considered.

One *horsepower* is 550 foot-pounds of work in one second, or 33,000 foot-pounds in one minute, or 1,980,000 foot-pounds in one hour.

The work necessary to be done in raising a body weighing W pounds through a height of h feet, equals Wh foot-pounds. The total work which any moving body is capable of doing in being brought to rest, equals its kinetic energy or $\frac{W\ v^2}{2\ g}$, when v is the velocity in feet per second.

Thus, the work which a cannon ball weighing 800 pounds and traveling with a velocity of 1,200 feet per second could do, is, $\frac{800 \times 1{,}200^2}{2 \times 32.16} = 17{,}910{,}447$ foot-pounds.

If stopped in one minute the horsepower would be 17,910,447 ÷ 33,000 = 542.8 nearly.

Force of a Blow.

In order to determine the force of a blow, the velocity of the object at the instant of striking must be known, and also the time required to bring the body to rest. It is a very difficult matter to determine the exact time, but a close approximation to the striking force may be obtained by dividing the kinetic energy of the body at the instant of striking by the average amount of penetration or compression produced by the striking body.

Let F = Striking force in pounds.
W = Weight of striking body in pounds.
v = Velocity of striking body in feet per second.
R = Distance penetrated or amount of compression = the distance through which the resistance acts, in feet..
t = Time required to bring the body to rest.
h = Height in feet which would produce the velocity v.

$$\text{Then, } F = \frac{W\,v}{g\,t} \text{ or } F = \frac{W\,v^2}{2\,g\,R} = \frac{W\,h}{R}.$$

For example, a steam hammer weighing 1,000 pounds (with its piston), falls from a height of eight feet and compresses a piece of iron one eighth of an inch, what is its striking force? If gravity be considered as the only force acting, the steam on top of the piston being used to prevent a rebound of the hammer, $F = \frac{W\,h}{R} = \frac{1{,}000 \times 8}{(\frac{1}{8} \div 12)} = 1{,}000 \times 8 \times 8 \times 12 = 768{,}000$ pounds. Divide one-eighth of an inch by twelve to obtain the amount of compression in feet or parts of a foot.

Safety Valves.

Balance the valve and lever over a sharp knife-like edge, and measure the distance from the point of suspension to the fulcrum (center of pin on which the lever turns).

Let a = Distance thus measured in inches.
b = Distance from center of valve to fulcrum in inches.
x = Distance of weight from fulcrum in inches.
W = Weight in pounds hung on lever.
Q = Weight of lever and valve in pounds.
A = Area of safety valves in square inches.
p = Pressure per square inch in the boiler.

$$\text{Then, } x = \frac{A\,p\,b - Q\,a}{W}.$$

$$W = \frac{A\,p\,b - Q\,a}{x}$$

$$p = \frac{W\,x + Q\,a}{A\,b}.$$

Gearing.

The *circular pitch* of a gear wheel is the distance in inches measured on the pitch circle from the center of one tooth to the center of the next tooth.

If the distance of the teeth of a gear thus measured were 2⅛ inches, we would say that the circular pitch was 2⅛ inches.

Let P = Circular pitch.
D = Diameter of pitch circle, in inches.
C = Circumference of pitch circle, in inches.
N = Number of teeth.
π = 3.1416.

$$P = \frac{C}{N} \text{ or } \frac{\pi D}{N}. \quad N = \frac{C}{P} \text{ or } \frac{\pi D}{P}$$

$$C = P N \text{ or } \pi D. \quad D = \frac{P N}{\pi} \text{ or } \frac{C}{\pi}.$$

Addendum = .3 P. Root = .4 P.

Thickness of teeth for cut gear = .5 P; for cast gear .48 P.

The *diametral pitch* of a gear wheel is the number of teeth in the wheel divided by the diameter of the pitch circle in inches, or, it is the number of teeth on the circumference of the gear wheel for one inch diameter of pitch circle.

A gear with a pitch diameter of 5 inches and having 40 teeth, is 8 pitch; one with the same pitch diameter and having 70 teeth is 14 pitch.

In the gear of 8 pitch there are 8 teeth on the circumference for each inch of the diameter of the pitch circle; and in one of 14 pitch there are 14 teeth on the circumference for each inch of the diameter of the pitch circle.

Let P = Diametral pitch.
D = Diameter of pitch circle, in inches.
N = Number of teeth.
d = Outside diameter.
l = Length of tooth.
t = Thickness of tooth.

$$P = \frac{N}{D}. \quad D = \frac{N}{P}. \quad N = P D. \quad d = \frac{N+2}{P}. \quad l = \frac{2.157}{P}. \quad t = \frac{1.57}{P}.$$

The circular pitch corresponding to any diametral pitch may be found by dividing 3.1416 by the diametral pitch; and the

diametral pitch corresponding to any circular pitch may be found by dividing 3.1416 by the circular pitch.

(*a*) If the diametral pitch of a gear is 6, what is the corresponding circular pitch?

(*b*) If the circular pitch is 1.5708 in., what is the corresponding diametral pitch?

(*a*) $\frac{3.1416}{6} = .5236$ in. (*b*) $\frac{3.1416}{1.5708} = 2$.

DIAMETRAL PITCHES WITH THEIR CORRESPONDING CIRCULAR PITCHES.

Diametral Pitch, or Teeth per inch in Diameter.	*Corresponding Circular Pitch.*	*Diametral Pitch, or Teeth per Inch in Diameter.*	*Corresponding Circular Pitch.*
1	3.1416	8	.3927
2	1.5708	9	.3491
3	1.0472	10	.3142
4	.7854	12	.2618
5	.6283	14	.2244
6	.5236	16	.1963
7	.4488	20	.1571

Belting.

D = Diameter of larger pulley in inches.
d = Diameter of smaller pulley in inches.
N = Revolutions per minute of larger pulley.
n = Revolutions per minute of smaller pulley.
W = Width of double belt in inches.
w = Width of single belt in inches.
H = Horsepower that can be transmitted by the belt.

Then, $H = \frac{D\ N\ w}{2750}$ for single belts.

$H = \frac{D\ N\ W}{1925}$ for double belts.

$w = \frac{2750\ H}{D\ N} = \frac{2750\ H}{d\ n}$; $W = \frac{1925\ H}{D\ N} = \frac{1925\ H}{d\ n}$.

$$D = \frac{2750\ H}{w\ N} \text{ for single belt.}$$

$$D = \frac{1925\ H}{W\ N} \text{ for double belt.}$$

$$N = \frac{2750\ H}{w\ D} \text{ for single belt.}$$

$$N = \frac{1925\ H}{W\ D} \text{ for double belt.}$$

The above rules are for open belts and pulleys having the same diameter, the arc of contact being, in this case, half the circumference, or 180°. For open belts and pulleys of different diameters, the arc of contact is less than 180° on the smaller pulley, and a different constant, to be taken from the following table, must be substituted in the formulas. To find the arc of contact, let l be the distance in inches between the centers of the pulleys. Then $\frac{D-d}{2\ l}$ = cosine of half the angle. Find this half angle from a table of natural cosines, and multiply by 2. The result is the arc of contact in degrees. Find the number in the first column of the table which is nearest to this result, and use the constant corresponding to that number. If a table of natural cosines is not at hand, measure the length of the arc of contact on the smaller pulley and divide it by the circumference of the pulley. Find the fraction in the second column which corresponds nearest to this result, and opposite this, its corresponding constant.

Degrees.	*Fraction of Circumference.*	*Single Belt Constant.*	*Double Belt Constant.*
90	$\frac{1}{4}$ = .25	6080	4250
112½	$\frac{5}{16}$ = .3125	4730	3310
120	$\frac{1}{3}$ = .3333	4400	3080
135	$\frac{3}{8}$ = .375	3850	2700
150	$\frac{5}{12}$ = .4167	3410	2390
157½	$\frac{7}{16}$ = .4375	3230	2250
180 to 270	$\frac{1}{2}$ to $\frac{3}{4}$ = .5 to .75	2750	1925

For example, what must be the width of a single belt to transmit 12 horsepower, when the diameter of the large pulley is 42 inches, of the smaller pulley 20 inches; distance between their centers 14 feet = 168 inches, and R. P. M. of smaller pulley 150?

$\frac{42-20}{2\times168}$ = .06548 = cosine of half the arc of contact, which thus = 86° 15′ nearly; 86° 15′ x 2 = 172½° = the arc of contact; the nearest number in the table is 180° and the corresponding constant is 2750; hence, $w = \frac{2750 \times 12}{20 \times 150} = 11$ inches.

Oak tanned leather makes the best belts. When belts are run with the hair side over the pulley, they have greater adhesion.

The ordinary thickness of leather belts is $\frac{1}{8}$ in., and weight about 60 lbs. per cubic foot.

Best vulcanized rubber belts are stronger than leather belts, their resistance being from 50 to 85 per cent. greater.

Rules for Calculating the Speed of Gears or Pulleys.

In calculating for gears, multiply or divide by the number of teeth, as may be required. In calculating for pulleys, multiply or divide by their diameter in inches.

The driving wheel is called the *driver*, and the driven wheel the *driven*.

Problem I.

The revolutions of driver and driven, and the diameter of driven being given, required the diameter of driver.

Rule.—Multiply the diameter of driven by its number of revolutions, and divide by the number of revolutions of the driver.

Problem II.

The diameter and revolutions of the driver being given, required the diameter of the driven to make a given number of revolutions in the same time.

Rule.—Multiply the diameter of the driver by its number of revolutions, and divide the product by the required number of revolutions.

Problem III.

The diameter or number of teeth, and number of revolutions of the driver, with the diameter or number of teeth of the driver being given, required the revolutions of the driven.

Rule. — Multiply the diameter or number of teeth of the driver by its number of revolutions, and divide by the diameter or number of teeth of the driven.

Problem IV.

The diameter of driver and driven, and the number of revolutions of driven being given, required the number of revolutions of the driver.

Rule.—Multiply the diameter of driven by its number of revolutions, and divide by the diameter of the driver.

HORSEPOWER OF ENGINES, BOILERS AND PUMPS.

Theoretical Horsepower.

The theoretical horsepower of any machine which uses a fluid (steam, gas, water, etc.) as a motive power or which discharges a fluid (*i. e.*, a pump or a fan) may be readily computed by the following formula, in which v is the volume of the fluid in cu. ft. per min., and p the average pressure in lbs. per sq. in.

$$\text{H. P.} = \frac{144\,v\,p}{33000}.$$

If, in the above formula, allowance for friction, etc., be made, the final result will be the actual horsepower.

Example.—A ventilating fan delivers 5.000 cu. ft. of air per minute at a pressure of .56 lb above the atmospheric pressure; what is the theoretical horsepower required to drive the machine?

$$\text{H. P.} = \frac{144\,v\,p}{33000} = \frac{144 \times 5.000 \times .56}{33000} = 12.218.$$

If all hurtful resistances be taken in this case as 20% of the total horsepower, the actual horsepower will be $12.218 \div (1 - .20) = 12.218 \div .80 = 15.2725$ H. P.

EXAMPLE.—The mean effective pressure computed from an indicator card taken from the air cylinder of an air compressor is 30.6 lbs. per sq. in.; diameter of cylinder, 28 in.; stroke, 48 in.; number of strokes per min., 108; what is the horsepower?

In this case

$$v = \frac{28^2 \times .7854 \times 48 \times 108}{1728} \text{ cu. ft.}$$

Hence, $\frac{144\,v\,p}{33000} = \frac{144 \times 28^2 \times .7854 \times 48 \times 108 \times 30.6}{1728 \times 33000} = 246.66$ H. P.

Horsepower of an Engine.

Let P = Mean effective pressure in pounds per square inch on the piston during one stroke.

L = Length of stroke in feet.

A = Area of piston in square inches.

N = Number of strokes per minute.

D = Diameter of piston in inches.

Then, to find the indicated horsepower,

$$\text{I. H. P.} = \frac{PLAN}{33000} = \frac{.238\ P\ L\ D^2\ N}{10{,}000{,}000}$$

The actual horsepower may be taken as three-fourths of the indicated horsepower. The mean effective pressure may be found exactly by taking some indicator cards, finding the areas by means of a planimeter, and dividing the area by the length of the card. Multiply the result by the scale of the indicator

ing, and the result is the mean effective pressure or M. E. P no planimeter is at hand, divide the card into ten equal parts d measure each part in the the middle as shown by the dotted es in the above figure.

Add all of the dotted ordinates together, and divide by ten; this result multiplied by the scale of the indicator spring, gives the M. E. P.

For example, a double-acting engine 26x30, making 80 revolutions per minute (80 R. P. M.), gives an indicator card, which being divided up and measured as shown in the figure, gives, for the total length of the ordinates, 21.4 inches. This divided by 10 equals 2.14 inches for the length of the mean ordinate. If a No. 40 spring is used in the indicator, every inch measured vertically on the diagram = 40 pounds per square inch, and 2.14x 40 = 85.6 pounds per square inch for the M. E. P. on the piston. Then the indicated horsepower or I. H. P. =

$$\frac{\text{P L A N}}{33000} = \frac{85.6 \times (\frac{30}{12}) \times (.7854 \times 26^2) \times (2 \times 80)}{33000} = 550.88.$$

The calculation is rendered much easier by using the second formula. Thus, I. H. P. =

$$\frac{238 \times 85.6 \times \frac{30}{12} \times 26^2 \times (2 \times 80)}{10{,}000{,}000} = 550.88.$$

If an indicator card cannot be obtained, a fair approximation to the M. E. P. may be obtained by adding 14.7 to the gauge pressure and multiplying the number opposite the fraction indicating the point of cut-off in the following table by the boiler pressure. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good simple non-condensing engines. If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke. Thus, if the stroke is 30 inches and the steam is cut off when the piston has traveled 20 inches, the engine cuts off at $\frac{20}{30} = \frac{2}{3}$ stroke. For a $\frac{3}{8}$ cut-off. and 92 pounds gauge pressure in the boiler, the M. E. P. is $[(92 + 14.7) \times .917 - 17] .9 = 72.76$ pounds per square inch.

Cut-off.	*Constant.*	*Cut-off.*	*Constant.*	*Cut-off.*	*Constant.*
$\frac{1}{8}$	.566	$\frac{5}{16}$	.771	$\frac{2}{3}$	.917
$\frac{1}{4}$	.603	$\frac{3}{8}$	.789	$\frac{7}{8}$	.926
$\frac{1}{4}$	.659	$\frac{1}{2}$	.847	$\frac{3}{4}$	.937
$\frac{3}{8}$	.708	$\frac{5}{8}$	.895	$\frac{4}{5}$	.944
$\frac{1}{16}$	.743	$\frac{5}{6}$	.904	$\frac{7}{8}$	.951

Horsepower of Boilers.

In actual practice the result of a great many tests has shown that an evaporation of 30 pounds of water per hour from a feed water temperature of 100° F. into steam at 70 pounds gauge pressure is the equivalent of one horsepower, or that this steam in a properly designed engine will do the equivalent of $33,000 \times 60 = 1,980,000$ foot-pounds of work per hour. In order, however, to have a more ready standard of comparison, the above evaporation has been reduced to another standard, and is found to be equal to the evaporation of 34.5 pounds of water from and at a temperature of 212° F. under atmospheric pressure, and it is upon this latter quantity that the calculations of the horsepower of boilers are usually based.

In making an approximation of the horsepower of a given boiler, the square feet of water heating surface of the boiler should first be determined, and in doing this the area of all the surfaces exposed to the fire and hot gases, which, on their opposite sides come in contact with the water in the boiler, should be taken into account.

EXAMPLE.—An externally fired flue boiler, having a shell 38 inches in diameter, and containing two flue pipes 10 inches in diameter, is 22 feet long without the smoke box. If the greatest depth of the water in the boiler is $\frac{2}{3} \times 38 = 25.33$ inches, what is the total water heating area of the boiler?

Six feet of the circumference of the boiler shell lies below the water line, as could be found by actual measurement, and the circumference of the two flues is—

$\left(\frac{10 \times 3.1416}{12}\right) \times 2 = 5.24$ feet, therefore the water heating surface of the shell equals—

$6 \times 22 = 132$ square feet, the water heating surface of the flues equals—

$5.24 \times 22 = 115.28$ square feet, and the water heating surface of the heads of the shell, that is, the area below the water line minus the area of the flues, which could be obtained by direct measurement, equals—

$4.5 \times 2 = 9$ square feet. Therefore, the total water heating surface of the boiler equals the sum of all these, or—

256.28 square feet.

Having determined the water heating surface of a boiler, to approximate its horsepower:

Rule.—Divide the total water heating surface in square feet by the number of square feet of heating area, as given in the table below, required to produce an evaporation equivalent to one horsepower in boilers of the given type.

EXAMPLE.—The total water heating surface of the above externally fired flue boiler is 256.28 square feet. What is the horsepower of the boiler?

By referring to the table we find that it takes about ten square feet of heating surface to produce a horsepower; therefore, the above boiler would be rated at about—

$$\frac{256.28}{10} = 25.63 \text{ horsepower.}$$

Type of Boiler.	*Square feet of water heating surface to produce one horsepower.*	*Ratio of water heating area to grate area required.*
Cylindrical.....	9	From 12 to 15 : 1
Flue	10	" 20 to 25 : 1
Fire-box Tubular.....	12	" 25 to 35 : 1
Return Tubular	15	" 25 to 35 : 1
Vertical.....	15	" 25 to 30 : 1
Water Tube.....	11	" 35 to 40 : 1

The above rule must not be taken as furnishing anything but an approximate method, since the same boiler will give a different horsepower whenever the conditions under which it is operated are changed; or, in other words, the horsepower developed depends largely upon the amount of coal burned per square foot of grate area per hour, the velocity and character of the furnace draft and the quality of the coal used. In ordinary practice, however, we may expect an evaporation of from 8 to 11 pounds of water from and at 212° F. for each pound of good

the projected area of the half-cylinder (or the diameter times the length of the cylinder) multiplied by the depth of the center of gravity of the half-cylinder, multiplied by the weight of a cubic inch of water, plus the diameter of the shell, multiplied by the pressure per square inch multiplied by the length of the cylinder.

If d = the diameter and l = the length of the cylinder, the pressure due to the weight of the water, when the cylinder is horizontal, upon the half-cylinder = $d \times l \times \frac{d}{2} \times$ the weight of a cubic inch of water $= l \times \frac{d^2}{2} \times$ the weight of a cubic inch of water d and l are to be measured in inches.

The pressure in pounds per sq. in., due to a head of water is equal to the head in feet multiplied by .434.

The head equals the pressure in lbs. per sq. in. multiplied by 2.304.

EXAMPLE.—(*a*) What is the pressure per sq. in. corresponding to a head of water of 175 ft.? (*b*) If the pressure had been 90 lbs. per sq. in., what would the head have been?

(*a*) $175 \times .434 = 75.95$ lbs. per sq. in.

(*b*) $90 \times 2.304 = 207.36$ ft.

Hydrokinetics.

Hydrokinetics, also called hydrodynamics and hydraulics, treats of water in motion. The velocity is not the same at all points of the flow, unless all cross-sections of the pipe or canal are equal. That velocity, which being multiplied by the area of the cross-section of the stream, will equal the total quantity discharged, is called the *mean velocity*.

Let Q = The quantity which passes any section in one second.
A = The area of the section.
v = The mean velocity in feet per second.

$$\text{Then } Q = Av, \text{ and } v = \frac{Q}{A}.$$

The vertical distance between the level surface of the water and the center of the aperture through which it flows, is called the head.

Let V = Mean velocity of efflux through a small aperture.
h = Head in feet at the centre of the aperture.
w = Weight of water flowing through the aperture per second, then, $V = \sqrt{2gh}$; that is, the velocity of efflux is the same as if the same weight of water had fallen through a height equal to its head.
Q = Theoretical number of cubic feet discharged per second.
V_m = Mean velocity through orifice in feet per second.
A = Area of orifice.
h = Theoretical head necessary to give a mean velocity V_m.
Q_a = Actual quantity discharged in cubic feet per second.

Then, for an orifice in a thin plate, or a square-edged orifice (the hole itself may be of any shape, triangular, square, circular, etc., but the edges must not be rounded), the actual quantity discharged is $Q_a = .615\ Q = .615\ AV_m$.

The *weir* is a device used for measuring the discharge of water. It is a rectangular orifice through which the water flows.

If d = the depth of the opening in feet, and b its breadth in feet, the area of the opening is $A = d \times b$, and the theoretical discharge is $Q = d \times b \times V_m = db \times \frac{2}{3}\sqrt{2gd}$, the head for this case being taken as d.

The actual discharge when the top of the weir lies at the surface of the water is $Q_a = .615\ Q = .615 \times db \times \frac{2}{3}\sqrt{2gd} = .615 \times \frac{2}{3} b \sqrt{2gd^3}$. If h_1 is the depth in feet of the top of a weir below the surface of the water, and h is the depth in feet of the bottom of the weir below the surface of the water, the actual discharge Q_a, in cubic feet per second is

$$Q_a = .615 \times \frac{2}{3} b \sqrt{2g}\left[\sqrt{h^3} - \sqrt{h_1^3}\right].$$

FLOW OF WATER IN PIPES.

Let V_m = Mean velocity of discharge in feet per second.
h = Total head in feet = the vertical distance between the level of the water in the reservoir and the point of discharge.
l = Length of pipe in feet.
d = Diameter of pipe in inches.
f = Coefficient of friction.

Then for straight cylindrical pipes of uniform diameter, the mean velocity of efflux may be calculated by the formula

$$V_m = 2.315 \sqrt{\frac{hd}{fl + .125d}}.$$

NOTE.—The head is always taken as the vertical distance between the point of discharge and the level of the water at the source, or point from which it is taken, and is always measured in feet. It matters not how long the pipe is, whether vertical or inclined, whether straight or curved, nor whether any part of the pipe goes below the level of the point of discharge or not, the head is always measured as stated above.

EXAMPLE.—What is the mean velocity of efflux from a 6-inch pipe, 5,780 feet long, if the head is 170 feet? Take f = .021

$$V_m = 2.315 \sqrt{\frac{h\,d}{f\,l + .125d}} = 2.315 \sqrt{\frac{170 \times 6}{.021 \times 5780 + (.125 \times 6)}}$$

= 6.69 feet per second.

When the pipe is very long compared with the diameter as in the above example, the following formula may be used, $V_m = 2.315 \sqrt{\frac{h\,d}{f\,l}}$, in which the letters have the same meaning as the preceding formula. This formula may be used when the length of the pipe exceeds 10,000 times its diameter.

The actual head necessary to produce a certain velocity V_m, may be calculated by the formula $h = \frac{f\,l\,V_m^2}{5.36\,d} + .0233\,V_m^2$.

If the head, the length of the pipe, and the diameter of the pipe are given, to find the discharge, use the formula, $Q = .09445\ d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}}$; that is, the discharge in gallons per seconds equals .09445 times the square of the diameter of the pipe in inches, multiplied by the square root of the head in feet times the diameter of the pipe in inches, divided by the coefficient of friction times the length of the pipe in feet, plus .125 times the diameter of the pipe in inches.

To find the value of f, calculate V_m by formula, $V_m = 2.315 \times \sqrt{\frac{h\,d}{f\,l}}$, assuming that $f = .025$, and get the final value of f from the following table:

$V_m =$	0.1	0.2	0.3	0.4	0.5	0.6
f =	.0686	.0527	.0457	.0415	.0387	.0365
$V_m =$	0.7	0.8	0.9	1	1¼	1½
f =	.0349	.0336	.0325	.0315	.0297	.0281
$V_m =$	2	3	4	6	8	12
f =	.0265	.0243	.0230	.0214	.0205	.0193

EXAMPLE.—The length of a pipe is 6,270 feet, its diameter is 8 inches, and the total head at the point of discharge is 215 feet. How many gallons are discharged per minute?

$$V_m = 2.315 \sqrt{\frac{215 \times 8}{.025 \times 6270}} = 7.67 \text{ feet per second, nearly.}$$

Using the value of $f = .0205$ for $V_m = 8$ (see table), $Q = .0955 \times 8^2 \sqrt{\frac{215 \times 8}{.0205 \times 6270 + (.125 \times 8)}} = 22.03$ gallons per second = $22.03 \times 60 = 1321.8$ gallons per minute.

If it is desired to find the head necessary to give a discharge of a certain number of gallons per second through a pipe whose length and diameter are known, calculate the mean velocity of efflux by using the formula $V_m = \frac{24.51\ Q}{d^2}$; find the value of f from the table, corresponding to this valve of V_m and substitute these values of f and V_m in the formula for the head.

For example, a 4-inch pipe, 2,000 feet long, is to discharge 24,000 gallons of water per hour; what head is necessary?

$\frac{24,000}{60 \times 60} = 6\frac{2}{3}$ gallons per second. $V_m = \frac{24.51 \times 6\frac{2}{3}}{4^2} = 10.2$ feet per second.

From the table, $f = .0205$ for $V_m = 8$, and .0198 for $V_m = 12$; assume that $f = 0.2$ for $V_m = 10.2$. Then $h = \frac{.02 \times 2000 \times 10.2^2}{5.36 \times 4} + .0233 \times 10.2^2 = 196.53$ feet.

To find the diameter of a pipe which will give any required discharge in gallons per second, the total length of the pipe and the head being known, use the following formula, which will give values accurate enough for practical purposes:

$$d = 1.229 \sqrt[5]{\frac{l\ Q^2}{h}}.$$

EXAMPLE.—What should be the diameter of a pipe in order to discharge 36,000 gallons of water per hour under a head of 50 feet, the length of the pipe to be 1,500 feet?

36,000 gallons per hour $= \frac{36,000}{60 \times 60} = 10$ gal. per sec. Hence,

$$d = 1.229 \sqrt[5]{\frac{1,500 \times 10^2}{50}} = 6.095'', \text{ say } 6''.$$

HEAT.

COEFFICIENT OF EXPANSION FOR A NUMBER OF SUBSTANCES.

Name of Substance.	*Linear Expansion.*	*Surface Expansion.*	*Cubic Expansion.*
Cast Iron	.00000617	.00001234	.00001850
Copper	.00000955	.00001910	.00002864
Brass	.00001037	.00002074	.00003112
Silver	.00000690	.00001390	.00002070
Bar Iron	.00000686	.00001372	.00002058
Steel (untempered)	.00000599	.00001198	.00001798
Steel (tempered)	.00000702	.00001404	.00002106
Zinc	.00001634	.00003268	.00004903
Tin	.00001410	.00002820	.00003229
Mercury	.00003334	.00006668	.00010010
Alcohol	.00019259	.00038518	.00057778
Gases	. . . . .	. . . . .	.00203252

A wrought iron bar 22 feet long is heated from 70° to 300°. How much will it lengthen ?

$22 \times (300 - 70) \times .00000686 = .0347116$ ft. $= .41654$ in.

Table of Specific Heats.

SOLIDS.

Copper	0.0951	Cast Iron	0.1298
Gold	0.0324	Lead	0.0314
Wrought Iron	0.1138	Platinum	0.0324
Steel (soft)	0.1165	Silver	0.0570
Steel (hard)	0.1175	Tin	0.0562
Zinc	0.0956	Ice	0.5040
Brass	0.0939	Sulphur	0.2026
Glass	0.1937	Charcoal	0.2410

LIQUIDS.

Water	1.0000	Lead (melted)	0.0402
Alcohol	0.7000	Sulphur "	0.2340
Mercury	0.0333	Tin "	0.0637
Benzine	0.4500	Sulphuric Acid	0.3350
Glycerine	0.5550	Oil of Turpentine	0.4260

GASES.

Air	0.23751	Superheated Steam	0.4805
Oxygen	0.21751	Carbonic Oxide	0.2479
Nitrogen	0.24380	Car' onic Acid	0.2170
Hydrogen	3.40900	Olefiant Gas	0.4040

The following table gives the temperatures and latent heats of fusion and of vaporization of a number of substances:

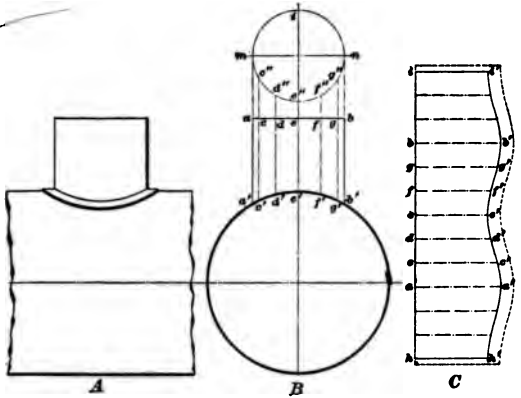
Substance.	*Temperature of Fusion.*	*Temperature of Vaporization.*	*Latent Heat of Fusion.*	*Latent Heat of Vaporization.*
Water	32°	212°	142.65	966.6
Mercury	−37.8°	662°	5.09	157
Sulphur	228.3°	824°	13.26	
Tin	446°		25.65	
Lead	626°		9.67	
Zinc	680°	1900°	50.63	493
Alcohol	Unknown	173°		372
Oil of Turpentine	14°	313°		124
Linseed Oil		600°		
Aluminum	1400°			
Copper	2100°			
Cast Iron	2192°	3300°		
Wrought Iron	2912°	5000°		
Steel	2520°			
Platinum	3632°			
Iridium	4892°			

How many units of heat are required to melt 10 pounds of zinc from a temperature of 60° F? The specific heat of zinc from the preceding table is .0956. Hence, the number of heat units necessary to raise it to the melting point is $10 \times (680 - 60) \times .0956 = 592.72$.

Latent heat of fusion = 50.63 heat units. Hence, the total number of heat units required is $592.72 + 10 \times 50,63 = 1099.02$.

To Develop the Dome of a Boiler.

A side view of the dome, together with a section of the boiler, is shown in figure *A*. Draw Fig. *B*, the end view of the dome and of the boiler. Above the dome draw a circle *i n e'' m*, of



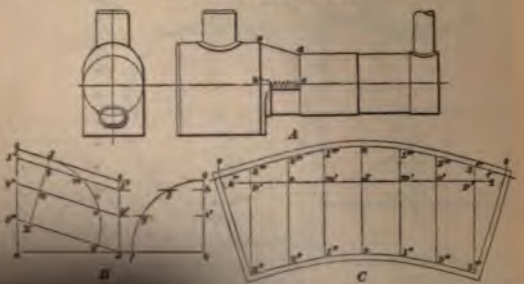
DRAWING.

the same diameter as the dome. Divide the lower half of this circle, as *n e'' m*, into any number of equal parts, as *m c''*, *c'' d''*, *d'' e''*, *e'' f''* and *f'' g''*. The greater the number of these divisions the more accurate will be the results. From the points of divisions *c''*, *d''*, *e''*, *f''* and *g''*, draw lines parallel to the vertical center line of the boiler as *c'' c'*, *d'' d'*, *f'' f'* and *g'' g'*.

We are now ready to draw the template of the dome as shown in Fig. *C*. Draw a straight line of indefinite length, and on it lay off a distance *h i* equal to the circumference of the dome. (The circumference of the dome is found by multiplying the diameter *a b* of the dome by 3.1416.) Divide the distance *h i* into twice the number of equal parts that the semicircle above the dome in Fig. *B* has been divided into. In the figure it has been divided into six equal parts, therefore, divide this line into 2 × 6

= 12 equal parts also, as $b\,g$, $g\,f$, $f\,e$, $e\,d$, etc., and through these points of division draw lines at right angles to the line $h\,i$, as shown; make the length of each of these lines the same as the length of the line which corresponds to it in Fig. *B*. Thus $e\,e'$ is equal to $e\,e'$ in Fig. *B*, $d\,d'$ is equal to $d\,d'$ in Fig. *B*. $a\,a'$ is equal to $a\,a'$ in Fig. *B*, etc. After having laid off the lengths of these lines, draw the curved line $i'\,e'\,h'$. This being done, we have the template of the dome on the seam. The lap for riveting must be allowed as shown by the dotted lines around the template.

To Develop the Slope Sheet a b c d of a Boiler Shown at A, in the Figure Below.



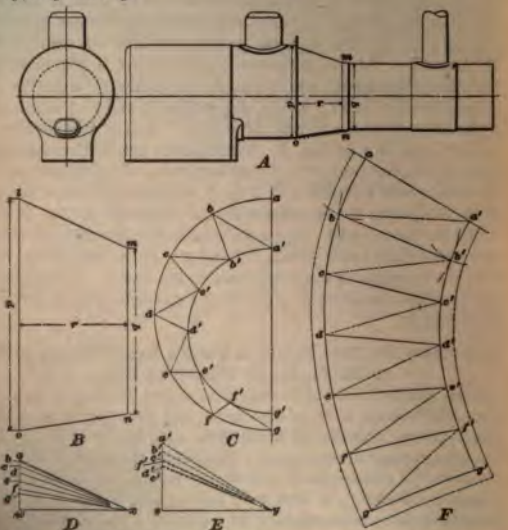
Draw a straight line a δ, as shown in Fig. *B*, and on it lay off [illegible] to *b c*, Fig. *A*. At *a* and *d* erect perpen[illegible]ectively, making *a c* equal to *b a*, and [illegible]*d*. With a point *b* on *a b* as a center, [illegible] the quadrant *f g*. Divide this quad[illegible]r parts, the greater the number the [illegible] results. Here it is divided into three, [illegible]rough the points *g*, *1* and *2*, draw lines [illegible]cting the perpendicular *d e* in *e*, *1'* and [illegible]icular *b g* in *h* and *i'*. Through the points [illegible] lines parallel to *c e*. Through any point as *J* [illegible]raw *J K* perpendicular to *c e*. From the line

J K lay off the distances *i m*, *n o* and *K p*, equal to the distances *h 1*, *i 2* and *b f*, respectively, and pass the dotted curve *J m o p* through the points. Now draw Fig. *C*. Draw the straight line *k q* and through the point *J*, draw *e c* perpendicular to it. Lay off on the line *k q* on each side of the line *c e*, points *m'* and *m'* at distances from it equal to the length of *J m* in Fig. *B*. Lay off also points *o* and *o'* at distances from *m'* and *m'* equal to *m o* in Fig. *B*; also, points *p'* and *p'* at distances from *o'* and *o'* equal to *o p* of Fig. *B*. Through the points thus laid off, draw lines parallel to *c e*. Lay off the points *c* and *e* from *J*, in Fig. *C*, equal to *J c* and *J e*, respectively, in Fig. *B*; the points *1'''* and *1''* from *m'* equal to *i 1''* and *i 1'* in Fig. *B*; points *2'''* and *2''* from *o'* and *o* equal to *n 2''* and *n 2'*, and the points *3'''* and *3''* from *p'* equal to *K 3''* and *K p* of Fig. *B*. Through the points thus laid off draw the curved lines *3''' c 3'''* and *3'' e 3''*. With the points *3''* as centers and a radius *a d*, Fig. *B*, describe the arcs *r* and *r*. With the points *3'''* as centers and a radius *3'' a*, Fig. *B*, describe the arcs *s* and *s*. From the points of intersection of these arcs draw lines to the points *3'''* and *3''*. This being done, we have the template of the slope sheet on the seams. The laps for riveting must be allowed as shown by the dotted lines around the template.

To Develop the Slope Sheet 1 m n o of a Boiler, Shown at A, in the Figure on Following Page.

Draw the two views of the sheet as shown in Figs. *B* and *C*. Suppose the seam to be at *o n* Fig. *A*, and the sheet to be made in one piece. Divide the semicircles *a d g* and *a' d' g'* Fig. *C*, into any number of equal parts; the greater the number of these divisions the more accurate will be the results. Join the points *b* and *b'*, *c* and *c'*, *d* and *d'*, *e* and *e'*, and *f* and *f'* by full lines, and join the points *b* and *a'*, *c* and *b'*, *d* and *c'*, *e* and *d'*, *f* and *e'*, and *g* and *f'* by dotted lines, as shown. Then draw Figs. *D* and *E*. Draw at right angles to one another the lines *w a* and *w x*, also the lines *z a'* and *z y*. Make the length of the line *w x* equal to *r* Fig. *B*, and the length of the line *w a* equal to *a a'* Fig. *C*. From *w* lay off on the line *w a*, Fig. *D*, distances *w b*, *w c*, *w d*, *w e*, *w f* and *w g*, respectively equal to the lengths of the full lines, *b b'*, *c c'*, etc., of Fig. *C*, and draw the

lines $a\ x$, $b\ x$, $c\ x$, $d\ x$, $e\ x$, $f\ x$ and $g\ x$, as shown. Make the length of the line $z\ y$, Fig. *E*, the same as that of $w\ x$, Fig. *D*. From z lay off on the line $z\ a'$, Fig. *E*, distances $z\ a'$, $z\ b'$, $z\ c'$, $z\ d'$, $z\ e'$ and $z\ f'$, respectively equal to the lengths of the dotted lines, $b\ a'$, $c\ b'$, etc., in Fig. *C*, and draw the lines $a'\ y$, $b'\ y$, $c'\ y$, $f'\ y$, $d'\ y$ and $e'\ y$.



We are now ready to draw the template of the slope sheet. Instead of drawing the whole template, we will draw only one-half of it as is shown in Fig. *F*, since the other half is exactly the same. Draw the line $a\ a'$, and make it equal in length to the distance $a\ x$, Fig. *D*. With a' as a center, and a radius $y\ a'$, describe an arc at b. With a as a center, and a radius $\ldots$, describe another arc intersecting the first arc in b.

With a' as a center, and a radius $w\ a$, Fig. *D*, describe an arc at b'. With b as a center, and a radius $x\ a$, Fig. *D*, describe an arc, intersecting the arc already drawn, at b'; draw the full line $b\ b'$ and dotted line $b\ a'$. With b' as a center, and a radius $y\ b'$, Fig. *E*, describe an arc at c. With b as a center, and a radius $z\ b'$, Fig. *E*, describe an arc cutting the last arc at c. With b' as a center, and a radius $w\ b$, Fig. *D*, describe an arc at c'. With c as a center, and a radius $x\ b$, Fig. *D*, describe an arc cutting the last arc at c'; draw the full line $c\ c'$ and dotted line $c\ b'$.

Continue to construct the remaining portion of the half template in a similar manner, taking the distances for the full lines from Fig. *D*, and those for the dotted lines from Fig. *E* Through the points a, b, c, d, e, f and g, and through the points a', b', c', d', e', f' and g' draw the curved lines shown. Since this is the development of the slope sheet at the seam, the laps for riveting must be allowed; they are shown by the dotted lines around the template in Fig. *F*.

MAKING BLUE PRINTS.

Blue print paper for copying tracings of plans and other drawings may be prepared as follows:

Dissolve 1 ounce avoirdupois, of ammonio-citrate of iron in 6 ounces of water, and in a separate bottle dissolve the same quantity of potassium ferricyanide in 6 ounces of water. Keep these solutions separate, and in a dark place or in opaque bottles.

To prepare the paper, mix equal quantities of the two solutions, and with a sponge spread it evenly over the surface. Let the paper remain in a horizontal position until the chemical has set on the surface, which will take but a few minutes, then hang the paper up to dry. In preparing the paper the room should be darkened by pulling down the shades, as direct rays of light affect the sensitized surface. The prepared paper should be kept in a closed drawer well covered with heavy paper so that no light can come in contact with the sensitized surface, otherwise it will lose much of its value.

To make a blue print of a tracing, lay the tracing with ink side down against the glass of the printing frame, then take the prepared paper and place the sensitized surface down on the

tracing. On the top of the paper place the felt cushion, on top of which place the hinged back of the printing frame, after which expose to the sunlight. The exposure will vary in sunlight from about 3 to 10 minutes. After the exposure, wash the paper thoroughly in a trough of cold water for about 10 minutes, and hang it up to dry.

The print after washing should be of a deep blue color with clear white lines. If the color is a pale blue, it signifies that it has not had sufficient exposure, and if the lines of the drawing are not perfectly clear and white, the exposure has been too long.

Corrections may be made on the print with an ordinary writing or ruling pen and a solution of washing soda, caustic potash, strong ammonia or any other alkali. When any of these are mixed with carmine ink the marks on the print will be red, thus distinguishing the corrections from the original.

BOILERS.

Points to be Observed in the Care and Inspection of Boilers.

Preliminary to a boiler inspection, the boiler, flues, mud drum, ash pit, and all connections should be thoroughly cleaned to facilitate a careful examination. Blisters may occur in the best iron or steel, and their presence, and also that of thin places, is ascertained by going over all parts of the boiler with a hammer. When blisters are discovered, the plates should be repaired or replaced. Repairing a blister consists in cutting out the blistered space and riveting a "hard patch" over the hole on the inside of the boiler, if possible, to avoid forming a pocket for sediment. All seams, heads, and tube ends should be examined for leaks, cracks, corrosions, pitting and grooving, detection of the latter possibly requiring the use of a magnifying glass. Uniform corrosion is a wasting away of the plates, and its depth can be determined only by drilling through the plate and measuring the thickness, afterwards plugging the hole. Pitting is due to a local chemical action, and is readily perceived. Grooving is usually due to buckling of the plates when under pressure, and frequently to the careless use of the sharp caulking tool. Seam leaks are generally caused by overheating, and demand

careful examination, as there may be cracks under the rivet heads. If such cracks are discovered the seams should be cut out, and a patch riveted on. Loose rivets should be carefully looked for, and should be cut out and replaced if found. Pockets, or bulging, and burns should be looked for in the fire box. The former are not necessarily dangerous, but if there are indications of their increasing, they should be heated and forced back into place or cut out and a patch put on. Burns are due to low water, the presence of scales, or to the continuous action of flames formed on account of air leaking through the brickwork. The burned spots should be cut out and patched as previously described. The condition of all stays, braces and their fastenings should be examined and defective ones replaced. The shell of the boiler should be thoroughly examined externally for evidences of corrosion, which is liable to set in on account of dampness, exposure to weather, leakage, etc., and may be serious. The boiler should be so set that joints and seams are accessible for inspection, and should have as little brickwork in contact with it as possible. The brickwork should be in good condition, and not have air holes in it, since they decrease the efficiency of the boiler and are liable to cause injury to the plates by burning as above explained, and also by unevenly heating and distorting them. The mud drum and its connections are liable to corrosion, pitting and grooving, and should be examined as carefully as the boiler.

All valves about a boiler should be easy of access, and should be kept clean and working freely. Each boiler should have at least three gauge cocks, properly located, and it is of the utmost importance that they be kept clean and in order, and the same may be said of the glass water gauge. The middle gauge cock should be at the water level of the boiler, and the other two should be placed one above and one below it, at a distance of about 6 inches.

The condition of the pumps or injectors should be looked into to make sure that they are in the best working order. The steam gauge should be tested to ascertain that it indicates correctly, and if it does not, it should be corrected. If the hydraulic test is to be used, the boiler should be tested to a

pressure of 50 per cent higher than that at which the safety valve will be set.

EXTERNAL INSPECTION WHEN BOILER IS UNDER STEAM.

The gauge cocks, and also the gauge glass, should be tried to make sure that they are not choked. The steam gauge should be taken down, if permissible, and tested, and corrected if necessary. The gauge pointer should move freely. Blowing out the gauge connection will show whether it is clear or not. The boiler connections should be examined for leaks. The safety valve should be lifted from its seat to make sure that it does not stick from any cause, and it should be seen that the weight is in the right place. Observe from the steam gauge if the valve blows off at the pressure it is set for. See that all pumps and feed apparatus are working properly, and that the blow-off and check valves are in order. Blisters and bagging may sometimes be detected in the furnace. The condition of the brickwork is of considerable importance, since the existence of air holes is a source of trouble, as explained.

One of the chief difficulties met with in boiler management is caused by incrustation, which is a deposit of mineral matters introduced in solution in the feed water, and slowly precipitated in the boiler by the action of heat. It is deposited on the boiler plates as a fine powder, but becomes hard and tenacious after a time under the influence of heat, and has to be scraped or chipped off. Incrustation or scale involves a decrease in the economy of operating a boiler, since it hinders the passage of heat considerably, and consequently necessitates the use of more fuel; under these conditions there is great danger of overheating the plates. Corrosion of the metal is also more likely to occur on those plates where the deposits of scale form, and some kinds of scale, notably sulphate of lime, cause corrosion on account of their chemical composition. The mineral substances most commonly found in feed waters are carbonate and sulphate of lime. Sometimes only one of these is present, but they are frequently found together. To prevent the formation of the carbonate scale, caustic soda or lime is usually employed, their effect being to precipitate lime or lime carbonate,

which settles, and may be blown out. The carbonate may be precipitated by heating the feed water; this prevents it from entering the boiler. For the sulphate of lime, the best remedy is carbonate of soda, but this must not be introduced in excess, as it is liable to cause foaming. The carbonate of soda precipitates the carbonate of lime, which can be blown out. When the carbonate and the sulphate occur together, carbonate of soda is still the best remedy. These scale preventatives are introduced into the boiler with the feed water. When possible, the feed water should be purified before entering the boiler; if the water is muddy or has much sediment, a filter is indispensable. Whenever very hard water has to be used, frequent blowing out of the mud drum is necessary to prevent the hardening of the precipitated matter. The presence of acids in boiler water is very detrimental to the metal of the boiler, since they cause corrosion, and they should be neutralized by some alkali, such as caustic soda or soda ash. Acid water may be detected by testing it with litmus paper; if the water turns blue litmus red, it is acid, but if it turns red litmus blue, it is alkali. Water is hard or soft according to the amount of mineral salts held in solution. Water is tested for hardness by mixing it with a soap solution; if the mixture has a very milky appearance, it indicates hardness. The soap solution may be prepared by dissolving scrapings of castile soap in alcohol, the two being mixed and allowed to stand a few days, being frequently shaken.

A precipitate caused by adding oxalic acid to a sample of feed water indicates the presence of lime.

Probably all natural alkali waters contain carbonate of lime.

To detect sulphate of lime in water, mix a little of the water with a solution of barium chloride, which any chemist can furnish. If this forms a white precipitate which will not redissolve on the addition of a little nitric acid, sulphate of lime must be present.

Great care should be taken to prevent the introduction of animal or vegetable oils into a boiler, since they cause mixtures and sometimes chemical combinations with the pulverized carbonate

of lime and magnesia. These collect in a mass at the bottom of the boiler and exclude water from the plates, causing a burning and bulging of the sheets.

Foaming or priming is usually due either to forcing a boiler beyond its capacity for furnishing dry steam, or to the presence of foreign matter. It is dangerous if occurring to any great extent, since water may be carried along with steam into the engine, and a cylinder head knocked out. Foaming, when it cannot be checked by the use of the surface blow-out apparatus, may necessitate the emptying of the boiler which must then be filled with fresh water; this rids the boiler of the impurities which have collected during the operation of the boiler.

Chimneys.

The *chimney* serves the double purpose of creating a draft and carrying away obnoxious gases. The production of the draft depends upon the fact that the furnace gases (the products of combustion) passing up the chimney have a high temperature, and are consequently lighter than an equal volume of outside air at the ordinary temperature; that is, the pressure within the chimney is slightly less than the pressure of the outside air. Consequently, the air will flow from the place of higher pressure to the place of lower pressure, that is, into the chimney through the furnace.

Suppose, for example, the average temperature of the gases in a chimney 150 feet high is 500° F. A pound of the gases at 62° F. has a volume of 12.5 cu. feet.; its volume at 500° is, then, $\frac{(12.5 \times 500 + 460)}{62 + 460} = 23$ cubic feet. Therefore, a column of the gases 1 foot square and 150 feet long would weigh $\frac{150}{23} = 6.52$ pounds. A similar column of air at 62° F. would weigh $\frac{150}{13.14} = 11.42$ pounds, nearly. Hence the pressure of the *draft is* $11.42 - 6.52 = 4.9$ pounds per sq. ft. = .947 in. of water.

It is evident that the pressure of the draft depends upon the temperature of the furnace gases and the height of the chimney. The higher the chimney, the lower may be the temperature of the gases to produce the same draft, and the greater will be the economy of the furnace. In general, chimneys are not built much less than 100 feet in height.

The relation between the height of the chimney and the pressure of the draft in inches of water is given by the following formula:

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right),$$

where p is the draft in inches of water, H the height of the chimney in feet, and T_a and T_c the absolute temperatures of the outside air and of the chimney gases, respectively. Absolute temperatures are found by adding 460° F. to the ordinary temperatures.

EXAMPLE.—What draft pressure will be produced by a chimney 120 feet high, the temperature of the chimney gases being 600° F.; of the external air 60° F.?

SOLUTION.—By the formula we find,

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right) = 120\left(\frac{7.6}{460+60} - \frac{7.9}{460+600}\right) = .85$$

inches of water.

The draft pressures ordinarily produced by chimneys vary from 0 to 2 inches of water. A water gauge pressure of 1 inch is equivalent to .036085 pounds per square inch. Wood requires least draft, and the small sizes of anthracite coal the greatest draft. To successfully burn anthracite, slack or culm, a draft of 1¼ inches is necessary.

To find the height of chimney to give a specified draft pressure, the formula may be transformed.

$$H = \left(\frac{p}{\frac{7.6}{T_a} - \frac{7.9}{T_c}}\right).$$

3934954

EXAMPLE.—Required the height of the chimney to produce a draft of 1⅛ inches of water, the temperature of the gases and of the external air being respectively 550° and 62° F.

SOLUTION.—By the formula we find,

$$H = \left(\frac{p}{\frac{7.6}{T_a} - \frac{7.9}{T_c}}\right) = \frac{1.125}{\frac{7.6}{522} - \frac{7.9}{1010}} = 167 \text{ ft.}$$

The sizes of chimneys for boilers of various horsepowers are given in the following table:

SIZES OF CHIMNEYS AND HORSEPOWERS OF BOILERS.

Height of Chimney in Feet.											*Actual Area in Sq. Ft.*	*Side of Sq. In.*	*Diameter in In.*
50	60	70	80	90	100	110	125	150	175	200			
Commercial Horsepower.													
23	25	27									1.77	16	18
35	38	41									2.41	19	21
49	54	58	62								3.14	22	24
65	72	78	83								3.98	24	27
84	92	100	107	113							4.91	27	30
	115	125	133	141							5.94	30	33
	141	152	163	173	182						7.07	32	36
		183	196	208	219						8.30	35	39
		216	231	245	258	271					9.62	38	42
			311	330	348	365	389				12.57	43	48
			363	427	449	472	503	551			15.90	48	54
			505	539	565	593	632	692	748		19.64	54	60
				658	694	728	776	849	918	981	23.76	59	66
				792	835	876	934	1023	1105	1181	28.27	64	72
					995	1038	1107	1212	1310	1400	33.18	70	78
					1163	1214	1294	1418	1531	1637	38.48	75	84
					1344	1415	1496	1639	1770	1893	44.18	80	90
					1537	1616	1730	1876	2027	2167	50.27	86	96

EXAMPLE.—A round chimney 100 feet high is to be used for a battery of boilers of 550 horsepower. What should be the *internal diameter?*

SOLUTION.—Looking under column 100 in "Height of Chimney in Feet" the nearest horsepower is 565, and the diameter corresponding is 60 inches, which should be the internal diameter of the chimney.

Chimneys are usually built of brick, though in some cases iron stacks are preferred. The external diameter of the base should be $\frac{1}{10}$ of the height in order to provide stability. The taper of a chimney is from $\frac{1}{8}$ to $\frac{1}{4}$ inch to the foot on each side. The thickness of brickwork is usually 1 brick (8 or 9 inches) for 25 feet from the top, increasing $\frac{1}{2}$ brick for each 25 feet from the top downward. If the inside diameter is greater than 5 feet, the top length should be $1\frac{1}{2}$ bricks, and if under 3 feet, it may be $\frac{1}{2}$ brick in thickness for the first 10 feet. A round chimney is better than a square one, and a straight flue better than a tapering one. If the flue is tapering the area for calculation is measured at the top.

The flue through which the gases pass from the furnaces to the chimney should have an area equal to, or a little larger than the area of the chimney. Abrupt turns in the flue or contractions of its area should be carefully avoided, as they greatly retard the flow of the gases. Where one chimney serves several boilers, the branch flue from each furnace to the main flue must be somewhat larger than its proportionate part of the area of the main flue.

MACHINE TOOLS.

Speed of Cuts for Machine Tools.

Brass is turned at about the same high speeds as wood. For cutting cast or wrought iron, either in lathe, planer, shaper, or slotting machine, a surface speed of 20 feet per minute is a good average, and 30 feet per minute should not be exceeded. Machinery steel is cut at about 14 feet per minute; extra hard steel at 9 feet per minute.

The Morse Twist-drill and Machine Co. recommends the rates of speed of twist-drills given in the following table

Diameter of Drills.	*Revolution of Drills per Minute.*		
	Steel.	*Iron.*	*Brass.*
1/16	940	1,280	1,560
1/8	460	660	785
3/16	310	420	540
1/4	230	320	490
5/16	190	260	320
3/8	150	220	260
7/16	130	185	230
1/2	115	160	200
9/16	100	140	180
5/8	95	130	160
11/16	85	115	145
3/4	75	105	130
13/16	70	100	120
7/8	65	90	115
15/16	62	85	110
1	58	80	100
1 1/16	54	75	95
1 1/8	52	70	90
1 3/16	49	66	85
1 1/4	46	62	80
1 5/16	44	60	75
1 3/8	42	58	72
1 7/16	40	56	69
1 1/2	39	54	66
1 9/16	37	51	63
1 5/8	36	49	60
1 11/16	34	47	58
1 3/4	33	45	56
1 13/16	32	43	54
1 7/8	31	41	52
1 15/16	30	40	51
2	29	39	49

They also give the rate of feed for twist-drills as follows:

Diameter of drill in inches.....	1/8	1/4	3/8 1/2	3/4 1 1 1/2
Number of revolutions per inch depth of hole.....	125	125	120 to 140	1 inch feed per minute.

Speed of Emery Wheels.

The speed most strongly recommended by their manufacturers is a peripheral velocity of 5,500 feet per minute for all

sizes. All things being considered, it is stated that there is no advantage to be gained in exceeding this speed. If run much slower than this, the wear on the wheels is much greater in proportion to the work accomplished, and if run much faster, the wheel is likely to burst.

Speed of Grindstones.

Grindstones used for grinding machinists' tools are usually run so as to have a peripheral speed of about 900 feet per minute, and those used for grinding carpenters' tools at about 600 feet per minute. In regard to safety, it may be stated in general that with any size of grindstone having a compact and strong grain, a peripheral velocity of 2,800 feet per minute should not be exceeded.

Speed of Polishing Wheels.

Polishing wheels are run at about the following peripheral speeds :

Leather-covered wooden wheels.....7,000 feet per minute.
Walrus-hide wheels.....8,000 " " "
Rag-wheels.....7,000 " " "

Change Gears Required for Cutting Screw Threads.

The pitch of a single threaded screw is the distance between two adjacent threads, measured on a line parallel to the axis of the screw; or, in *any* screw, whether single or multiple threaded, it is the distance the nut is moved by one revolution of the screw. Usually, a screw is spoken of as having a certain number of threads to the inch, and this is equal to the number of revolutions the screw must make in order to move the nut a distance of one inch; so, whether the screw be single or multiple threaded, the pitch is always equal to 1 divided by the number of revolutions that the screw must make in order to move the nut one inch.

The Simple Geared Lathe.

In Fig. 1 is shown the usual arrangement of the change gears of a simple geared screw cutting lathe. By a simple geared lathe is meant a lathe in which the change gears are so arranged that the circumferential velocity of the change gear on

the stud is the same as that of the change gear on the lead screw, which means that, when the change gear on the stud has rotated, say, five teeth, the change gear on the lead screw has also rotated five teeth, whatever the diameter of these gears, or of any intermediate gears between them, may be.

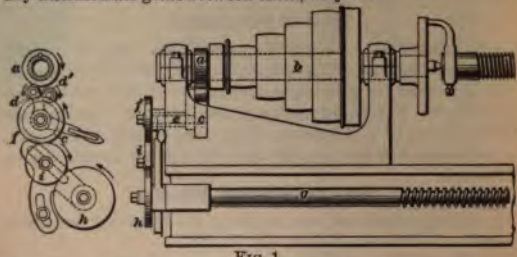


FIG 1.

Referring to Fig. 1, the gear *a* is fastened to the spindle *b* and drives another gear *c* by means of either one of the reversing gears *d*, *d'*. The gear *c* is keyed to one end of the spindle *e*; this spindle is called the *stud*, and carries on its outer end a change gear *f*. The lead screw *g* carries a change gear *h*; and these two change gears *f* and *h* are connected by means of the *idler* gear *i*, so that gear *f* drives gear *h*, and with it, the lead screw *g*.

In making calculations for the change gears of a simple geared screw cutting lathe, the idler gear *i* is ignored, as it is only introduced to connect gears *f* and *h*. The gears d and d' are also ignored, since they are only used to change the direction of rotation of the gear *c*, their duty being to facilitate the cutting of either right-hand or left-hand threads; when *d* meshes with gear *a*, as shown in Fig. 1, a right-hand thread is cut, and when d' meshes with gear *a*, a left-hand thread is cut.

The number of teeth in the gear *a* is not always the same as the number of teeth in the gear *c*; it is so in some lathes, but in others it is not; hence, in calculating the change gears for any lathe, the numbers of teeth in the gears *a'* and *c'* *must* be taken into account.

By the following formulas and rules, the numbers of teeth required in the change gears in order to cut a given number of threads to the inch, or the number of threads to the inch that given change gears will produce may be found.

Let a = the number of teeth in the spindle gear a.
c = the number of teeth in the gear c.
f = the number of teeth in the change gear on stud.
h = the number of teeth in the change gear on lead screw.
g = the number of threads to the inch in the lead screw.
n = the number of threads to the inch to be cut.

Then, $n = \frac{g c h}{a f}$ (1). $h = \frac{n a f}{g c}$ (3).

$\frac{h}{f} = \frac{n a}{g c}$ (2). $f = \frac{g c h}{n a}$ (4).

Now, of the gears h, f, c, a, a and f are the *drivers*, and c and h, being driven by a and f, are called the *driven* gears; remembering this, we deduce, from formula 1, the following rule for simple geared screw cutting lathes.

RULE.—*The number of threads to the inch to be cut is equal to the number of threads to the inch in the lead screw, multiplied by the product of the numbers of teeth in the driven gears, and divided by the product of the numbers of teeth in the driving gears.*

EXAMPLE.—If the lead screw, g, of a simple geared lathe has 5 threads to the inch, and the gear a has 21 teeth, the gear c, 42 teeth; the change gear f, 60 teeth; and the change gear h, 72 teeth; how many threads to the inch will be cut?

SOLUTION.—Using formula 1, we have,

$$n = \frac{g c h}{a f} = \frac{5 \times 42 \times 72}{21 \times 60} = 12 \text{ teeth.}$$

From formula 2, we deduce the following rule for simple geared screw cutting lathes.

RULE.—*The number of teeth in the change gear on the lead screw, divided by the number of teeth in the change gear on the stud, is equal to the product of the number of threads to the inch to be cut*

and the number of teeth in the driving spindle gear, divided by the product of the number of threads to the inch in lead screw and the number of teeth in the fixed gear on the stud.

EXAMPLE.—If the lead screw g, of a simple geared lathe has 8 threads to the inch, and the gear a has 16 teeth, and the gear c, 32 teeth, how many teeth must there be in each of the gears, f and h, in order that the lathe may cut 10 threads to the inch?

SOLUTION.—Using formula 2,

$$\frac{h}{f} = \frac{n\,a}{g\,c} = \frac{10 \times 16}{8 \times 32} = \frac{5}{8},$$

and, if it were possible to have gears with 5 and 8 teeth respectively, then a solution of the problem would be, $h = 5$; $f = 8$. It is evident that such gears are impracticable; but, as it does not change the value of a fraction to multiply both numerator and denominator by the same number, we may multiply 5 and 8, each by such a number that the resulting numbers of teeth in the gears are satisfactory. There is evidently, therefore, *more than one solution* to the problem—for if we multiply by 10 we shall have, $h = 50$; $f = 80$, which would give 12 threads to the inch; and if we multiply by 13, we shall have, as another solution, $h = 65$: $f = 104$, which would also give 12 threads to the inch, because $\frac{65}{104} = \frac{5}{8}$.

Having found that $\frac{h}{f} = \frac{5}{8}$, it is customary in practice to choose the change gears in the following manner. From the assortment of gears belonging to the lathe, choose one of convenient diameter, the number of whose teeth is divisible by either the numerator 5 or the denominator 8, and after dividing by one of these numbers, multiply both numerator and denominator by the quotient.

EXAMPLE.—Given $\frac{h}{f} = \frac{5}{8}$; to find the numbers of teeth in the change gears h and f, respectively.

SOLUTION.—Choose a gear of convenient diameter, the number of whose teeth, say 60, is divisible by either 5 or 8, in this case by 5; divide 60 by 5 and the answer is 12. Then, $\frac{5 \times 12}{8 \times 12} = \frac{60}{96}$; that is h has 60 teeth and f, 96 teeth.

If one of the change gears is given, and it is desired to find the number of teeth in the other change gear, in order to cut a given number of threads to the inch, use either formula 3 or formula 4, according as the number of teeth in gear *h* or in gear *f* is required. After the examples given, these formulas will not need explanation.

In a simple geared screw cutting lathe, it is often possible to cut a *fractional number of threads* to the inch, as is the case in the following example.

EXAMPLE.—If the lead screw *g* has two threads per inch, and the gear *a* has 20 teeth, and the gear *c* has 20 teeth, how many teeth must there be in each of the change gears *f* and *h*, in order to cut 5¼ threads to the inch?

SOLUTION.—Using formula 2,

$$\frac{h}{f} = \frac{n\,a}{g\,c} = \frac{5\frac{1}{4} \times 20}{2 \times 20} = \frac{5\frac{1}{4}}{2}.$$

Then, choosing a gear whose number of teeth, say 32, is divisible by 2, divide 32 by 2 and the quotient is 16. Then, $\frac{5\frac{1}{4} \times 16}{2 \times 16} = \frac{84}{32}$; that is, *h* has 84 teeth and *f*, 32 teeth. In many cases, however, it is impossible, out of the assortment of gears supplied with a simple geared screw cutting lathe, to find gears to cut a screw of the required number of threads to the inch. In such cases, it becomes necessary, either to *make* suitable gears or to resort to a compound geared lathe.

THE COMPOUND GEARED LATHE.

In Fig. 2 is shown the usual arrangement of the change gears of a compound geared screw cutting lathe. The difference between this and the simple geared lathe lies in putting two change gears of different sizes on one spindle, in place of the idler, between the gear on the stud and the gear on the lead screw. These two gears on one spindle are shown at *i* and *j* in Fig. 2, the gear *j* meshing with gear *h* on the lead screw, and gear *i* meshing with gear *f* on the stud.

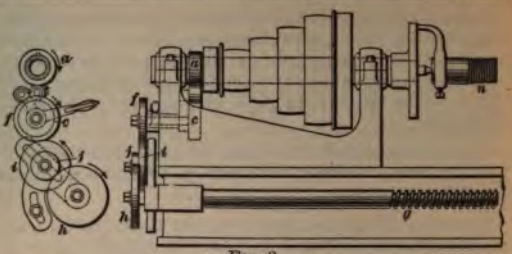


FIG. 2.

From the following formulas, the numbers of teeth in each change gear, or the number of threads per inch which can be cut with given change gears, can be found.

Let a = the number of teeth in the spindle gear a.

c = the number of teeth in the gear c.

f = the number of teeth in the change gear f.

h = the number of teeth in the change gear h.

i = the number of teeth in the change gear i which meshes with the change gear f.

j = the number of teeth in the change gear j which meshes with the change gear h.

g = the number of threads to the inch in the lead screw.

n = the number of threads to the inch to be cut.

$$\text{Then, } n = \frac{g \times c\,h\,i}{a\,f\,j} \qquad (5).$$

Now, remembering that gears a f and j are the drivers, and gears c, h and i are the *driven* gears, and also that the idlers are ignored in all calculations, we can, from formula 5, deduce the following rule for compound geared screw cutting lathes:

RULE.—*The number of threads to the inch to be cut is equal to the number of threads to the inch in the lead screw, multiplied by the product of the number of the teeth in each of the driven gears, and divided by the product of the number of teeth in each of the driving gears.*

EXAMPLE.—If the lead screw g of a compound geared lathe has 2 threads to the inch, and the gear a has 20 teeth; gear c, 40

teeth ; change gear *f*, 48 teeth ; change, gear *i*, 72 teeth ; change gear *j*, 36 teeth; and change gear *h*, 96 teeth, how many threads to the inch will be cut ?

SOLUTION. — Using formula 5, we have, $n = \frac{g \times c \, h \, i}{a \, f \, j} = \frac{2 \times 40 \times 96 \times 72}{20 \times 48 \times 36} = 16$ threads to the inch.

If it is desired to find what combination of change gears will enable us to cut a given number of threads to the inch, the following formula may be used,

$$\frac{i}{j} = \frac{n \, a \, f}{g \, c \, h} \qquad (6).$$

From this formula the following rule is deduced :

RULE.—*Of the change gears of a lathe, any driven gear divided by any driver gear is equal to the product of the numbers of teeth in each of the other driver gears and the number of threads to the inch to be cut, divided by the product of the numbers of teeth in each of the other driven gears and the number of threads to the inch in the lead screw.*

EXAMPLE.—In a compound geared lathe, in which the lead screw has 5 threads to the inch, gear *a*, 20 teeth, gear *c*, 40 teeth, and the number of threads per inch to be cut is 3½, what must be the numbers of teeth in each of the change gears *h*, *i*, *j*, *f* ?

SOLUTION.—Using formula 6, we have,

$$\frac{i}{j} = \frac{n \, a \, f}{g \, c \, h}.$$

From the assortment of gears belonging to the lathe, choose, for the driven gear *h*, one whose number of teeth, say 28, can be divided by the number of threads per inch to be cut, in this case 3½ ; 28 is a multiple of 3½ because it is obtained by multiplying 3½ by 8. Substitute this value in place of *h* ; then choose any gear of convenient size, say one having 40 teeth, and substitute 40 in place of *f* ; we shall then have,

$\frac{i}{j} = \frac{n \, a \times 40}{g \, c \times 28}$ or, substituting the given values of *n*, *a*, *g*,

and c, we have,

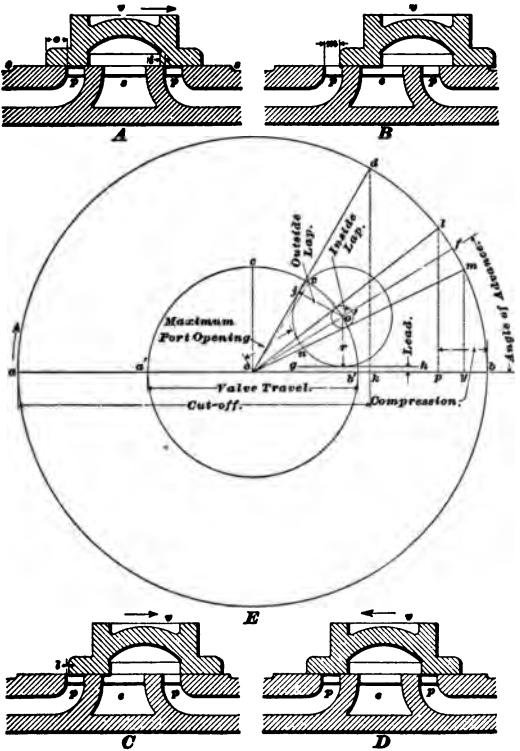
$$\frac{i}{j} = \frac{3\frac{1}{2} \times 20 \times 40}{5 \times 40 \times 28} = \frac{1}{2}.$$

Choose, for j, a gear whose number of teeth, say 60, is divisible by 2; then dividing the number of teeth in j by 2, we have $60 \div 2 = 30$. Now multiplying both terms of the fraction $\frac{1}{2}$ by 30, $\frac{i}{j} = \frac{1 \times 30}{2 \times 30} = \frac{30}{60}$; that is, $i = 30$ and $j = 60$. Hence, one solution of the problem is, $h = 28$; $i = 30$; $j = 60$; $f = 40$.

THE SLIDE VALVE.

Figs. *A*, *B*, *C* and *D* show sections of an ordinary **D** slide valve at different points of its travel. Fig. *A* shows the valve in its central position, with the center of the valve in line with the center line of the exhaust port. The names of the various parts are as follows: *p* and *p* are the *steam ports*; *e* is the *exhaust port*; *s s* is the *valve seat*; the amount, *o*, by which the valve overlaps the outer edges of the steam ports is the *outside lap*; the amount, *i*, by which the valve overlaps the inside edges of the steam port is the *inside lap*; the amount, *l*, (Fig. *C*,) which the port is open when the piston is at the end of the stroke is the *lead*. The valve travel is the total distance in one direction which the valve can be moved by the eccentric; it is the total distance between two extreme positions of the valve. The *displacement* of the valve is the distance which the valve has moved (in either direction) from its central position.

The line joining the center of the eccentric with the center of the crank shaft is called the *eccentric radius*. When the eccentric radius makes a right angle with the center line of the crank, that is, when the eccentric radius is vertical (see *o e*, Fig. *E*), the valve is in its central position, provided the valve seat is horizontal, as is usually the case. When the crank is on a dead center, say *a*, Fig. *E*, the valve must be in the position shown in Fig. *C*; that is, it must have moved from its central position an amount equal to the outside lap plus the lead. In order that this may happen the eccentric must be at *c*, Fig. *E*. The angle, *c o c*, *which the eccentric* must be moved from its vertical position *when crank is on a* dead center is called the angle of advance.


A
B
Outside Lap.
Inside Lap.
Maximum Port Opening.
Lead.
Angle of Advance.
Valve Travel.
Cut-off.
Compression.
E
C
D

In Fig. *B*, the valve is shown in its extreme position at the right. The distance marked *m* is the *maximum port opening*. It matters not whether the outer edge of the valve travels beyond the inner edge of the port or falls short of it as in the figure, the distance *m* between the edge of valve and the edge of the port when the valve is in its extreme position is the maximum port opening. If, in Fig. *C*, the valve were shown moving to the left, a little further movement would bring the left outer edge just even with the outer edge of the left steam port, and from here on to the end of the stroke no more steam could enter the left end of the cylinder; in other words, the valve *cuts off* at this point. A little further movement of the valve to the left brings the valve to the position shown in Fig. *D*, with the right inner edge opposite the inner edge of the right steam port; it is at this point that compression begins.

When designing a valve for an engine, some of the above quantities are assumed and the remaining ones are required; these may be found by means of the diagram shown in Fig. *E*.

Let *a b*, Fig. *E*, drawn to any convenient scale, represent the stroke of the engine, then *a d b* will represent the crank pin circle. About *o*, the center of the crank pin circle, describe a circle, *a' e b'*, whose diameter, *a' b'*, is equal to the actual travel of the valve. Draw the line *g h* parallel to *a b* and at a distance from it equal to the lead of the valve. Then, with a radius *o' j* equal to the outside lap of the valve, describe a circle, called the outside lap circle, tangent to the line *g h*, and having its center *o'* on the circle *a' e b'*. Draw the line *o o'*, and produce it to *f*; then $f\,o\,b = e\,o\,c =$ the *angle of advance*.

Now, draw any position of the crank center line, such as *a o*, and drop upon it, from the point *o'*, a perpendicular; the length of this perpendicular, marked *r*, in Fig. *E*, is the displacement of the valve for that position of crank center line. *r* is not marked correctly on the cut; it should extend to the line *a b*.

About the center *o'*, with a radius equal to the inside lap of the valve, describe a circle; this is called the inside lap circle.

The radius *o d*, drawn from the point *o* tangent to the outside lap circle, is the position of the center line of crank at the point of cut-off. Drop a perpendicular from point *d*, meeting the

line $a\ b$ at k; then $a\ k$ is the distance moved by piston before cut-off, and the fraction of the stroke at which the valve cuts off is represented by the fraction $\frac{a\ k}{a\ b}$.

Draw the radius $o\ l$ tangent to the upper side of the inside lap circle, and it will be the position of the center line of the crank when *compression* commences; if a perpendicular be dropped from point, l, meeting the line, $a\ b$, at p, the fraction of the stroke of piston, at which compression begins, will be represented by the fraction $\frac{a\ p}{a\ b}$.

In like manner, the radius, $o\ m$, drawn tangent to the lower side of the inside lap circle, is the position of the center line of the crank at the moment of *release*; and $\frac{a\ y}{a\ b}$ is the fractional part of the stroke at which the expanding steam is released.

The maximum steam port opening is equal to $o\ n$, n being the point of intersection of the outside lap circle with the angle of advance line, $o\ f$.

The essential features of the valve diagram having been given, the following examples will make clear its application in practice.

EXAMPLE 1.—Given the point of cut-off, the point of release, the lead, and the maximum port opening, to find the valve travel, the outside and inside lap, the angle of advance, and the point of compression.

SOLUTION.—Draw to a convenient scale the crank pin circle, $a\ c\ b$, Fig. *E*, having its center at o, and its diameter, $a\ b$, equal to the stroke of the piston.

From the point, a, lay off, on the line, $a\ b$, the distances $a\ k$ and $a\ y$, so that $\frac{a\ k}{a\ b}$ and $\frac{a\ y}{a\ b}$ are equal, respectively, to the fractions of the stroke at which cut-off and release are to occur. At

k and y draw perpendiculars to the line $a\ b$, intersecting the crank pin circle at d and m, respectively; the radii, $o\ d$ and $o\ m$, will represent the positions of the crank at cut-off and release, respectively. Now draw $g\ h$ parallel to $a\ b$, and at a distance above it equal to the lead; then, about o as a center, and with a radius equal to the given maximum port opening, describe an arc. Find by trial a center o', from which a circle can be drawn tangent to this arc, and also to the radius, $o\ d$, and to the line, $g\ h$. The radius of this circle will be the required outside lap; and its center o' will be a point in the valve circle whose center is at o; this circle can now be drawn, since the radius $o\ o'$ is known.

The diameter, $a'\ b'$, is equal to the required valve travel. Now, with o' as a center, draw a circle tangent to $o\ m$, and the radius of this circle will be the required inside lap. Draw $o\ f$ through o' and the angle $f\ o\ b$ is the required angle of advance. Draw the radius $o\ l$ tangent to the inside lap circle on its upper side, and $l\ p$ perpendicular to $a\ b$.

Then $\frac{a\ p}{a\ b}$ represents the fraction of the stroke at which compression begins.

EXAMPLE 2.—Given the valve travel, the angle of advance, the cut-off, and the point of compression, to find the lead, and the outside and inside lap.

SOLUTION.—Draw the crank pin circle, as before, and the valve circle $a'\ e\ b'$; construct the angle $f\ o\ b$ equal to the angle of advance. By the same method as employed in the last example locate the radii, $o\ d$ and $o\ l$, representing the positions of the crank at the points of cut-off and compression, respectively.

About the point, o', at which $o\ f$ intersects the valve circle, describe a circle tangent to $o\ d$, and the radius, $o'\ j$, of this circle will be the required outside lap. Now draw the line, $g\ h$, parallel to $a\ b$ and tangent to the outside lap circle; then, the perpendicular distance between $g\ h$ and $a\ b$, is the required lead. The radius of a circle drawn from o' tangent to $o\ l$ will be the inside lap.

EXAMPLE 3.—Given the valve travel, outside lap, and the lead, to find the point of cut-off and angle of advance.

SOLUTION.—Draw the crank pin circle and the valve circle $a' e b'$ as before; draw a line parallel to $a b$, at a distance above it equal to the outside lap r plus the lead, intersecting the valve circle at the point o'. About o' as center, and with a radius equal to the given lap, describe a circle; draw $o d$ tangent to this circle, and drop a perpendicular from d meeting line $a b$ at a point k; then the required cut-off is represented by the fraction $\frac{ak}{ab}$. Draw the radius of through the point o' and the angle $f o b$ is the required angle of advance.

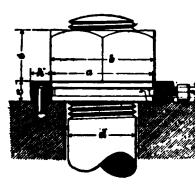
EXAMPLE 4.—Given the outside lap, the lead, and the point of cut-off; to find the valve travel and the angle of advance.

SOLUTION.—Draw the crank pin circle as before, and by the same method as employed in Example 1 locate the radius $o d$, the position of the crank at the point of cut-off. Draw $g h$ parallel to $a b$, and at a distance above it equal to the lead. At a distance above the line $a b$ equal to the lap plus the lead, draw another line parallel to $a b$; about a center o' on this line, and with a radius $o' j$ equal to the outside lap, describe a circle tangent to $o d$ and $g h$. Draw the radius of through o', then $f o b$ will be the required angle of advance. About o as a center, and with a radius oo', describe the valve circle $a' e b'$, and $a' b'$ will be the required valve travel.

MACHINE DESIGN.

Lock Nuts.

A good method of locking a nut is shown in the figure. The lower portion of the nut is turned down, and in the center of the circular portion a groove is cut. A collar is fastened by means of a pin to one of the pieces to be connected and into this collar is fitted the circular part of the nut. The nut is then bound to the collar by a set screw passing through the latter, the point of the set screw en-



gaging into the groove turned in the nut. The following proportions have proved very satisfactory, in which d, the diameter of the bolt is taken as a unit. All dimensions are in inches:

$a = 1\frac{1}{2}\,d - \frac{1}{16}''$ $\qquad$ $f = \frac{1}{2}\,d + \frac{1}{8}''$.
$b = 1\frac{1}{2}\,d + \frac{1}{8}''$. $\qquad$ $g = \frac{5}{8}\,d + \frac{1}{16}''$.
$c = \frac{1}{4}\,d + \frac{1}{4}''$. $\qquad$ $h = \frac{1}{4}\,d + \frac{1}{4}''$.
$e = \frac{3}{4}\,d$.

Proportion of Keys.

In common designing, the sizes of keys are determined by empirical formulas, which gives an excess of strength. For an ordinary sunk key, the following proportions may be adopted:

$b = \frac{1}{4}\,d$. $\qquad$ t = thickness of key in inches.
$t = \frac{2}{3}\,b = \frac{1}{6}\,d$. $\qquad$ b = breadth of key in inches.
d = diameter of shaft in inches.

Line Shafting.

The speed of a shaft is fixed largely by the speed of the driving belt or the diameters of the pulleys upon it. In general, machine shop shafts run about 120 to 150 revolutions per minute; shafts driving wood-working machinery, about 200 to 250 revolutions per minute; in cotton mills, the practice is to make the shaft diameter smaller and run at a higher speed. Line shafts should generally not be less than $1\frac{3}{4}$ inches in diameter.

The distance between the bearings should not be great enough to permit a deflection of more than $\frac{1}{100}''$ per foot of length. Hence, when the shaft is heavily loaded with pulleys, the bearings must be closer than when it carries only a few pulleys.

Below are given the maximum distances between the bearings of different sizes of continuous shafts which are used for the transmission of power.

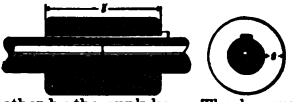
DISTANCES BETWEEN BEARINGS.

Diameter of Shaft in Inches.	*Distance Between Bearings in Feet.*	
	Wrought Iron Shaft.	*Steel Shaft.*
2	11	11.5
3	13	13.75
4	15	15.75
5	17	18.25
6	19	20
7	21	22.25
8	23	24
9	25	26

Pulleys which give out a large amount of power should be placed as near a hanger as possible.

Shaft Couplings.

A *box*, or *muff coupling*, is shown in the figure. It consists of a cast iron cylinder which fits over the ends of the shaft. The two ends are prevented from moving relatively to each other by the sunk key. The key-way is cut half into the box and half into the shaft ends. Quite commonly the ends of the shafts are enlarged to allow the key-way to be cut without weakening the shaft.



The key may be proportioned by the formula already given. For the other dimensions take

$$l = 2\tfrac{1}{2}\,d + 2''.$$
$$t = .4\,d + .5''.$$

EXAMPLE.—Find the dimensions of a muff coupling for a shaft $2\frac{1}{4}''$ in diameter.

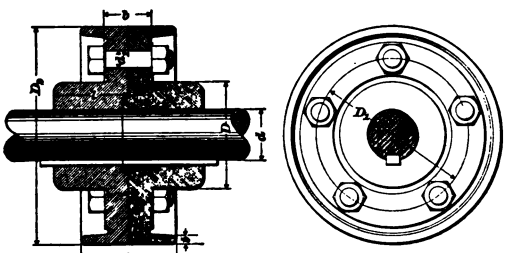
SOLUTION.—For the key we use the formula previously given,

$$b = \tfrac{1}{4}\,d = \tfrac{1}{4} \times 2\tfrac{1}{4} = \tfrac{9}{16}''.$$
$$t = \tfrac{1}{8}\,d = \tfrac{1}{8} \times 2\tfrac{1}{4} = \tfrac{9}{32}''.$$

For the muff,

$$l = 2\tfrac{1}{2}\,d + 2'' = 2\tfrac{1}{2} \times 2\tfrac{1}{4} + 2'' = 8\tfrac{1}{4}''.$$
$$t = .4\,d + .5'' = .4 \times 2\tfrac{1}{4} + .5'' = 1\tfrac{1}{4}''.$$

A *flange coupling* is shown in the following figure. Cast iron flanges are keyed to the ends of the shafts. To insure a perfect joint the flange is usually faced in the lathe after being keyed to the shaft. The two flanges are then brought face to face and bolted together.



Sometimes the ends of the shafts are enlarged to allow for the key-way. To prevent the possibility of the shafts getting out of line, the end of one may enter the flange of the other.

The [illegible] proportions may be used for this form of flange coupling:

d = diameter of shaft.

Diameter of boss $D = 2\frac{1}{2}\,d - 1''$.

Diameter of nut circle $D_1 = 2\frac{1}{2}\,d - 2''$.

$= 1\frac{1}{2}\,d - 1''$.

Number of bolts $= 4 - \frac{d}{2}$

Take the nearest whole number.

Diameter of bolts $d_1 = \frac{d}{4} - \frac{1}{8}$

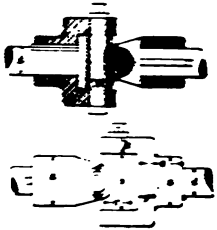
Diameter of coupling $D_2 = 1.6\,D$.

$= 2\,d - 2''$.

$t = \frac{1}{2}\,D$.

$t = 2\,d$.

The proportions for the key have already been given.



In the accompanying figure is shown a flexible coupling, or *universal joint*. These joints, when constructed of wrought iron, may have the following proportions:

d = diameter of shaft.

$a = 1.6\,d$. $e = 1.6\,d$.

$b = 2\,d$. $g = .6\,d$.

$c = d$. $h = .5\,d$.

$k = .6\,d$.

Pedestals.

The [illegible] pedestal, [illegible] box, *bearing* and *journal box* are [illegible] They are all a form of bearing, and [illegible] a bearing piece.

A form of [illegible] box frequently used for small shafts is [illegible] Fig. 1. It consists of two parts: 1, the box which [illegible] the journal, and 2, the cap which is screwed down to [illegible] In [illegible] the seats are of babbitt, or as it is [illegible] the box is babbitted. The cap is held in [illegible] caps or cheeks. This is invariably done in [illegible] pedestals.

The proportioning of a pedestal is largely a matter of experi[illegible] or some of the parts are calculated for strength.

All the proportions of the pedestals which follow are based on [illegible] of the journal d as the unit; the length of the [illegible] the same as that of the journal.

The journal box shown in Fig. 1, the following proportions

may be used for sizes of journals from ¼″ to 2″ diameter inclusive. The diameter of shaft d = the unit.

a = 2.25 d.	m = .25 d + .1875″.
b = 1.75 d.	n = .5 d.
c = d.	o = .625″ (constant).
e = .375 d.	p = 1.5 d.
f = .08 d + .0625″.	q = 1.333 d.

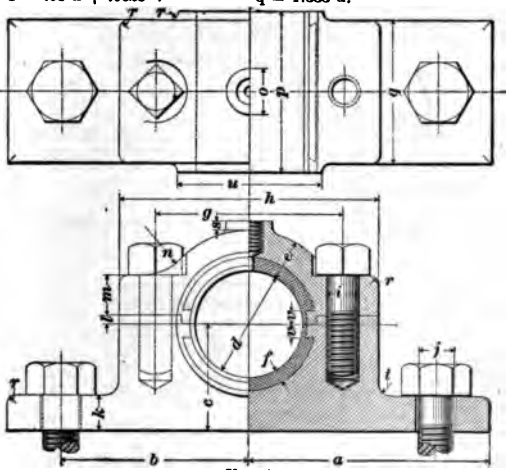


FIG. 1.

g = 1.75 d.	r = .08 d.
h = 2.45 d.	s = .125″ (constant).
i = .3 d.	t = .16 d.
j = .33 d.	u = 1.333 d.
k = .25 d + .125″.	v = .125 d.
l = .08 d.	

In Fig. 2 is shown a common form of pedestal, which is used for somewhat larger journals than the one shown in Fig. 1.

It consists of (1) a foundation plate which is bolted to the foundation on which the pedestal rests; the plate is essential when the pedestal rests on brickwork or masonry, but may be dispensed with when the pedestal rests on the frame of the machine; (2) the block which carries the seats and supports the journal; (3) the cap which is screwed down over the seats. The

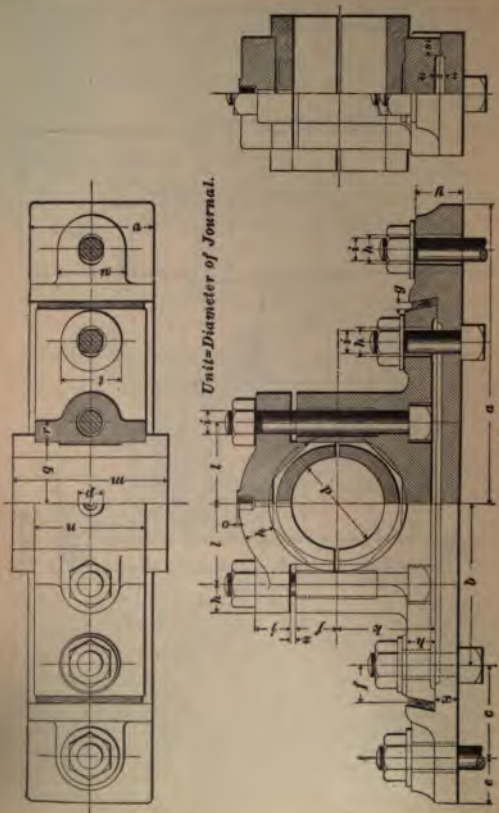


Fig. 2.

bolt holes in both foundation plate and block are oblong, so that the pedestal may be readily adjusted.

The following proportions may be used for this kind of pedestal, having journals from 2″ to 6″, inclusive. An oil cup having a ¼″ pipe tap shank may be used on pedestals for journals having diameters from 3″ to 4″, and ⅜″ pipe tap shank for larger sizes up to 6″ diameter.

NOTE.—The shanks of oil cups and grease cups bought in the market are made with a ⅛″, ¼″, ⅜″ or ½″ pipe thread. The amount of oil or grease the cup holds when filled is usually expressed in ounces

The diameter of journal d = the unit.

a = 3.25 d.	n = 1.25 d.
b = 1.75 d.	o = .125″ (constant).
c = d.	p = .875″ (constant).
e = .5 d.	q = .625 d.
f = .4375 d.	r = .25 d.
g = .09 d.	s = .1875 d.
h = .3125 d.	t = .65 d.
i = .25 d.	u = .75 d.
j = .375 d.	v = 1.375 d.
k = 1.0625 d.	x = .25 d.
l = .875 d.	y = .5 d.
m = 1.75 d.	z = .0625 d.

Fig. 3 shows a pedestal suitable for the crank shaft of a horizontal engine with journals from 8″ to 20″ in diameter. The block may be complete in itself, as shown in the figure, but more often it forms part of the engine bed.

The seats are in three parts, and may be adjusted horizontally by means of the wedges W. The lower seat may be raised by placing packing pieces under it. To obtain its dimensions, use the following proportions, which are based on the unit d = the diameter of the crank shaft journal.

a = d + 1″.	q′ = 1.5 d.
b = .5d + 1″	r = .15 d.
c = .66 d.	r′ = .1 d.
e = .825 d − 25″.	r_1 = d.
f = .6 d.	s = .9 d.
g = .1 d + .5625″.	t = .15 d + .375″.
h = .1 d + .25″.	t′ = .9 d.
h′ = .08 d.	u = 1.5 d.
i = .11 d.	v = .25 d + .375″.
j = .625″ (constant).	w = 1.45 d.
k = .5 d + 1.25″.	w′ = 1.47 d.
l = .375″ (constant).	w_1 = 1.75 d.
m = .175 d + .3125″	x = .1 d.
n = .25 d + .25″.	y = .3 d + .75″.
n′ = .1 d + .375″.	y_1 = .2 d + .5″.
o = 1″ (constant).	z = .09 d.
p = .25 d + .625″.	z′ = 2.5″ (constant.)
q = 1.75 d.	

Taper of adjusting wedge, 1 : 10.

Further details of the bottom seat and the cap are shown in

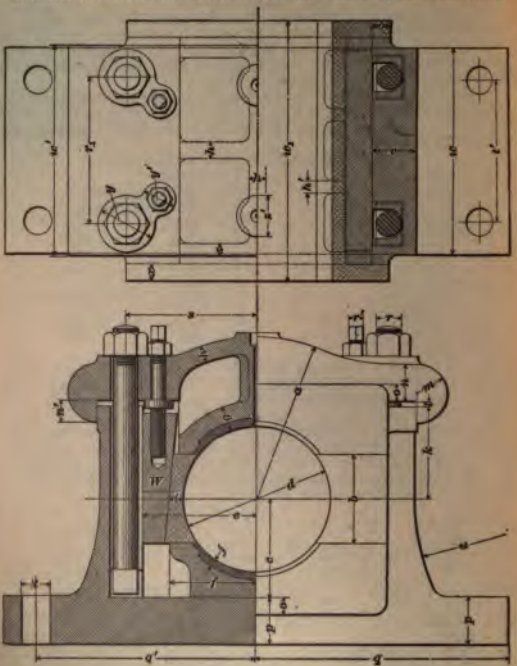


FIG. 3.

Fig. 4, in which the unit is the same as in Fig. 3, and the proportions are as follows:

a = 1° (constant). c = .08 d.
b = 1.65 d — .5". d = .1 d.

The foundation, or bed casting, is shown in Fig. 5, and has

dimensions to suit the pedestal shown in Fig. 3. Its proportions are as follows :

The diameter of the crank shaft journal d = unit.

a = 2.45 d + 7.25".
b = 2.3 d + 5.25".
c = .5 d + 3.5".
e = 3.5 d + 2".
f = .25 d + .5".
g = .25 d + 1.75".
h = .25 d + 2.25".
i = .05 d + .5"
j = .05 d + 1.125".
k = .05 d + .75".
l = .25 d + .75".
m = .4 d
m' = .6 d.
n = 1.55 d + 2.5".
o = .25 d + 2".
o' = .25 d + .5".
o" = .5 d + 4.5".
p = .08 d.
q = 1.5 d.
r = .15 d + .375".
s = .15 d + .375".
t = .9 d.
u = .15 d + .875".
v = .2 d.
w = 1.5 d.
x = 1.65 d.

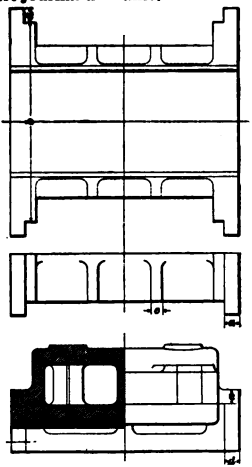
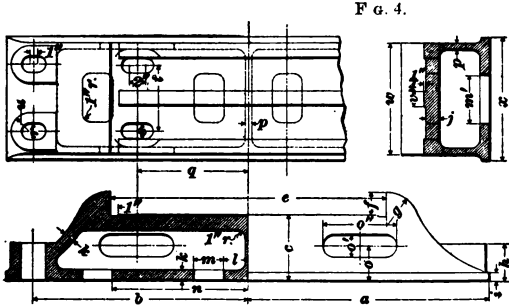


FIG. 4.



Hangers.

A hanger is used when a shaft bearing is to be suspended from the ceiling. The following figure shows a form of a hanger made by a leading manufacturing company.

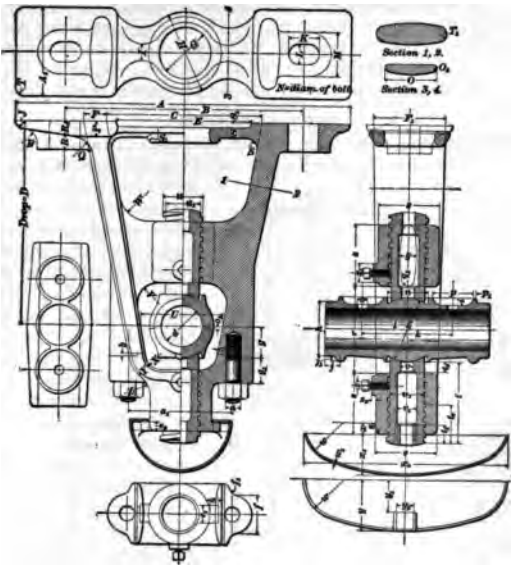
The frame of the hanger is divided and the parts are connected by bolts. With such a form, the shaft may be more easily removed than when the hanger frame is a solid piece.

The units for determining the leading dimensions of a shaft hanger are the diameter *d* of the shaft and the drop D of the hanger.

The following proportions are suitable for shafts ranging from 1¼″ to 4¼″ in diameter :

$A = 6\,d + .45\,D.$
$A_1 = 2\,d + .03\,D.$
$B = 4\,d + .35\,D.$
$C = 2\,d + .3\,D.$
$E = 2\,d + .25\,D.$
$F = .5\,d + .01\,D.$
$F_1 = 1.5\,d + .05\,D.$
$G = 1.25\,d.$
$H = .2\,d.$
$I = .4\,d.$
$J = .125\,d + .01\,D.$
$K = .5\,d + .5''$
$L = .25\,d + .5''.$
$M = .75\,d + .6875''.$
$N = .25\,d + .375''.$
$O = .125\,d.$
$O_1 = .094\,d + .002\,D.$
$P = .375\,d + .008\,D.$
$Q = .375\,d + .008\,D.$
R and R_1 (see note).
$S = .25\,d + .005\,D.$
$S_1 = .125\,d + .003\,D.$
$T = .125\,d + .01\,D.$
T_1 = (see note).
$U = 2\,d.$
$V = .5\,d.$
$W = .75\,d.$
$X = .375\,d.$
$Y = .25\,d + .125''.$
$Z = .625\,d.$
$a = .15\,d + .375''.$
$a_1 = 2.4\,d + .3125''$
$b = .08\,d.$
$c = .125\,d + .0625''.$

$e = .2\,d.$
$e_1 = .4\,d.$
$e_2 = .2\,d.$
$f = .375\,d + 1''.$
$f_1 = .09\,d + .25''.$
$g = .75\,d.$
$g_1 = 1.3125\,d + .125''.$
$h = 1.25\,d + .1875''.$
$i = .1\,d.$
$j = .25\,d + .25''.$
$j_1 = .125\,d + .0625''.$
$k = 2.2\,d.$
$l = 4\,d.$
$m = 1.4\,d + .375''.$
$n = d.$
$o = .25d.$
$o_1 = .0625\,d.$
$p = d.$
$p_1 = .0625\,d.$
$q = .4\,d.$
$q_1 = .15\,d.$
$r = 2.125\,d.$
$s = 1.5\,d.$
$s_1 = .125\,d.$
$t = 2\,d.$
$t_1 = .5\,d.$
$t_2 = d.$
$t_3 = .25\,d.$
$u = .95\,d.$
$u_1 = .85\,d.$
$v = .25\,d + .125''.$
$v_1 = .5\,d.$
$w = d.$
$w_1 = .125''$ (constant).



$x = .25\ d.$
$x_1 = d.$
$x_2 = 4\ d + 2''.$
$y = 1.25\ d.$
$y_1 = .75\ d + .0625''.$
$y_2 = .4\ d + .0625''.$
$z = .06\ d + .75''.$
$z_1 = .12\ d + .75''.$
$z_2 = .3125''$ (constant).

Thread of plugs, .5" pitch for all sizes.

NOTE.—To find R_1, draw the arc J; also, draw the arc Q tangent to P; then, draw a straight line tangent to these arcs, and R_1 will be the distance along the center line determined by B included between this tangent and the upper face of the hanger. Having found R_1, make R equal to it.

The radius T_1 is made equal to $\frac{1}{3}$ the thickness at the middle.

The steps of the ball and socket bearings are of cast iron and

PROPORTIONS OF PULLEYS—*Continued.*

Diam.	Face.	Rim.		Arm.		Hub.		Boss.		
		A	B	C	D	E	F	G	H	I
66	12	11/16	1/2	3 9/16	1 9/16	10	1 1/2	1	1 3/4	1/2
	16	....	....	....	....	11 1/2	1 5/8	1 1/4	2	....
	20	1/2	3/4	4 1/4	1 11/16	13 1/2	1 7/8	....	....	....
	24	....	....	....	....	15	....	....	....	....
72	12	3/8	5/16	3 7/8	1 1/16	10 1/2	1 5/8	1 1/4	2	1/2
	16	....	....	....	....	12 1/2	1 3/4	....	....	....
	20	9/16	1/16	4 5/8	2 1/16	13 1/2	1 7/8	....	....	....
	24	....	....	....	....	15	2	....	....	....

Rope Belting.

There is a growing tendency toward the substitution of hemp and cotton ropes for belting and line shafting as a means of transmitting power in large factories and shops. The advantages claimed for the rope driving system are:

1. Economy; for a rope system is cheaper to install than either leather belting or shafting.

2. In the rope system there is less loss of power by slipping.

3. Flexibility; that is, the ease with which the power is transmitted to any distance and in any direction.

In this country, a single rope is carried around the pulley as many times as is necessary to produce the required power, and the necessary tension is obtained by passing the rope around a tension pulley which is weighted to give the desired tension.

The ropes used in rope transmission are either of hemp, manilla or cotton. Manilla ropes are mostly used in this country. They are of three strands hawser laid, and may be from 1/2 inch to 2 inches in diameter.

The weight of ordinary manilla or cotton rope is about .3 D^2 pounds per foot of length, where D represents the diameter of the rope in inches. Letting w = the weight per foot of length,

$$w = .3\ D^2.$$

The breaking strength of the rope varies from 7,000 to 12,000 lb. per sq. in. of cross-section. The average value may be taken as 7,000 D^2, when D is the diameter of rope.

For a continuous transmission, it has been determined by experiment that the best results are obtained when the tension in the driving side of the rope is about $\frac{1}{35}$th of the breaking strength. That is,

$$T_1 = \text{tension in tight side} = \frac{7{,}000\ D^2}{35} = 200\ D^2.$$

The ropes run in V-shaped grooves, and the coefficient of friction is, of course, greater than on a smooth surface. The coefficient for grooves with sides at an angle of 45° may be taken at from .25 to .33.

The horsepower that can be transmitted by a single rope running under favorable conditions is given by the formula

$$H = \frac{v\ D^2}{825}\left(200 - \frac{v^2}{107.2}\right),$$

in which H = horsepower transmitted.

D = diameter of rope in inches.

v = velocity of rope in feet per second.

The maximum power is obtained at a speed of about 84 feet per second. For higher velocities, the centrifugal force becomes so great that the power is decreased, and when the speed reaches 145 feet per second, the centrifugal force just balances the tension so that no power at all is transmitted. Consequently, a rope should not run faster than about 5,000 feet per minute, and it is preferable on the score of durability to limit the velocity to 3,500 feet per minute.

EXAMPLE.—A rope fly-wheel is 26 feet in diameter and makes 55 revolutions per minute. The wheel is grooved for 35 turns of 1½ inch rope. What horsepower may be transmitted?

SOLUTION.—Velocity in feet per second =

$$v = \frac{26 \times \pi \times 55}{60} = \frac{4{,}492}{60} = 74.9 \text{ feet.}$$

Applying the formula

$$H = \frac{v\ D^2}{825}\left(200 - \frac{V^2}{107.2}\right) = \frac{74.9 \times (1\frac{1}{2})^2}{825}\left(200 - \frac{(74.9)^2}{107.2}\right) =$$

30.16, the horsepower transmitted by one rope or turn. Then

both, and on which the rope runs. For a rope composed of 7-wire strands, the pulley diameter should not be less than 150 times the rope diameter.

The horsepower transmitted by a wire rope is, approximately,

$$H = 3.1\ d^2\ v,$$

in which d = diameter of rope in inches, and v = velocity of rope in feet per second.

The deflection of the rope required to give the necessary tension may be found by the formula,

$$h = .0000695\ S^2,$$

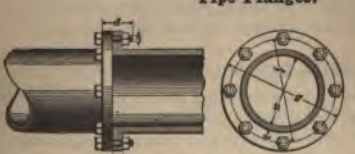
in which h = deflection of rope in feet when at rest, and S = span in feet.

EXAMPLE.—The velocity of a wire rope is 75 feet per second; (a) what should be the diameter of the rope in order to transmit 125 horsepower? (b) How much should the rope sag, the span being 160 feet?

SOLUTION.—(a) Substituting in the formula, $H = 3.1\ d^2\ v$, $.125 = 3.1 \times d^2 \times 75 = 232.5\ d^2$; whence, $d^2 = \frac{125}{232.5}$, and $d = \sqrt{\frac{125}{232.5}} = .73''$, say $\frac{3}{4}''$.

(b) Using the second formula given above, the sag, or deflection, $h = .0000695 \times 160^2 = 1.8$ feet.

Pipe Flanges.



The figure shows the method of flanging and bolting the ends of two cast iron pipes. The dimensions of the flanges for the various sizes of pipes are given in the following table:

STANDARD PIPE FLANGES.

n = number of bolts.

a	*b*	*c*	*d*	*n*	*e*	*f*	*g*
2.0	.409	5/8	2.0	4	5/8	4.75	6.0
2.5	.429	5/8	2.25	4	11/16	5.25	7.0
3.0	.448	5/8	2.5	4	3/4	6.0	7.5
3.5	.466	5/8	2.5	4	13/16	6.5	8.5
4.0	.486	3/4	2.75	4	15/16	7.25	9.0
4.5	.498	3/4	3.0	8	15/16	7.75	9.25
5	.525	3/4	3.0	8	15/16	8.5	10.0
6	.563	3/4	3.0	8	1	9.625	11.0
7	.600	3/4	3.25	8	1 1/16	10.75	12.5
8	.639	3/4	3.5	8	1 1/16	11.75	13.5
9	.678	3/4	3.5	12	1 1/16	13.0	15.0
10	.713	7/8	3.625	12	1 1/16	14.25	16.0
12	.790	7/8	3.75	12	1 1/4	16.5	19.0
14	.864	1	4.25	12	1 3/8	18.75	21.0
15	.904	1	4.25	16	1 3/8	20.0	22.25
16	.946	1	4.25	16	1 7/16	21.25	23.5
18	1.020	1 1/8	4.75	16	1 7/16	22.75	25.0
20	1.090	1 1/8	5.0	20	1 11/16	25.0	27.5
22	1.180	1 1/4	5.5	20	1 11/16	27.25	29.5
24	1.250	1 1/4	5.5	20	1 7/8	29.5	32.0
26	1.300	1 1/4	5.75	24	2	31.75	34.25
28	1.380	1 1/4	6.0	28	2 1/16	34.0	36.5
30	1.480	1 3/8	6.25	28	2 1/16	36.0	38.75
36	1.710	1 3/8	6.5	32	2 3/16	42.75	45.75
42	1.870	1 1/2	7.25	36	2 5/16	49.5	52.75
48	2.170	1 1/2	7.75	44	2 3/4	56.0	59.5

Cylinders and Steam Chests.

Fig. 1 shows a cylinder designed for a simple slide valve engine. The front head *A* is cast solid with the cylinder. The method of fastening to the frame *B* is clearly shown.

The principal dimensions of this cylinder may be determined from the following proportions:

D = diameter of cylinder.

L = length of stroke + thickness of piston + twice the piston clearance.

C = length of stroke + distance from outer edge to outer edge of piston rings — (.01 D + .125").

a = 5.5 i.

b = 4.2 i.

c = i.

d = i.

e' = the net area of a single cylinder head bolt whose nominal diameter is $e = \frac{A\,P}{4{,}000\,n}$,

where A = the area of cylinder head in square inches.

P = the steam pressure.

n = the number of bolts.

The pitch of the bolts may be from 4.5 to 5.5 inches, but should never be more than 5 f.

f = 1.5 i.

g = .04 D + .125". Take the nearest nominal size pipe tap.

h = twice the outside diameter of drain pipe.

i = .0003 PD + .375", where P is the steam pressure. If the steam pressure is less than 100 pounds make P = 100.

j = .85 i.

k = 4 i.

l = .75 i.

m = 1.01 D + .125".

n = m + 6 e, never less. Here e is the nominal diameter of the bolt.

o = the nominal diameter of steam chest bolts. The net area of a single steam chest bolt $= \frac{A'\,P}{4{,}000\,n'}$

where A' = area of steam chest.

n' = number of bolts in steam chest.

p = 2.75 o.

q = 1.5 r.

r = 1.25 i.

s = i. This is required only when the length of the port is greater than 12".

t = 1.25 i. When D is greater than 24" use four bolts in the standard and make t = 1.1 i.

u = 1.5 i.

v = .25", constant.

The dimensions of the *steam ports*, *exhaust ports*, and other steam passages depend upon the velocity of the flow of steam.

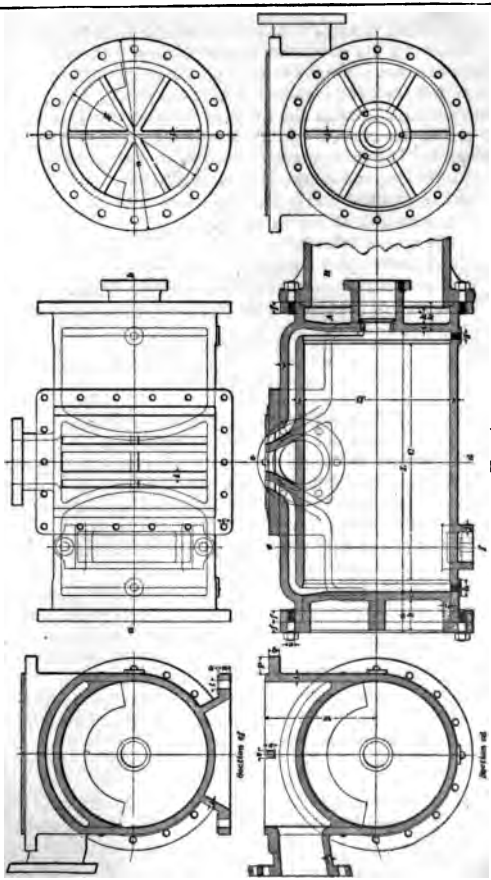


FIG. 1.

The ports and passages must be large enough to allow the steam to follow up the advancing piston without loss of pressure. The maximum allowable velocity of the steam in the passages, when they are short, is about 160 feet per second. But, with the ordinary ratio between the length of connecting rod and length of crank, the average velocity is about 5/8 of the maximum. Hence, the allowable average velocities are 100 to 125 feet per second for long and short passages, respectively.

Let l = length of port in inches.
b = breadth of port in inches.
A = area of cylinder.
S = average piston speed in feet per second.
v = average velocity of steam in feet per second.

Then, area of port × velocity of steam = area of piston × velocity of piston,

$$\text{or } l\,b\,v = A\,S,$$

$$\text{whence } l\,b = \frac{A\,S}{v}.$$

Take v = 100 for long indirect passages,
Take v = 125 for short direct passages.

The constant 100 may be used for v, when designing plain slide valve engines of the ordinary type which cut off late in the stroke, and 125 may be used for high speed engines with early cut-off, and for the Corliss type.

The area of the exhaust port or ports may be from 1 3/8 to 2 1/2 times the area of a steam port.

The area of the cross-section of the steam pipe is approximately equal to the area of the steam port; likewise, the area of the exhaust pipe should be equal to that of the exhaust port.

The length l of the port may be .6 D to .9 D for slide valve engines, and about .9 D to D for the Corliss type.

The height *w*, Fig. 1, of the valve seat must be such that the area of the most contracted part of the exhaust port is not less than 75% of the area of the steam port.

THE STEAM CHEST.

Fig. 2 shows a steam chest for the cylinder illustrated in Fig. 1. The principal dimensions are to be determined by the follow-

ing proportions, which are based upon the thickness i of the cylinder walls, and upon the travel and dimensions of the valve :

a = length of valve + travel of valve + twice the clearance between the valve and the steam chest at ends of valve travel.

b = breadth of valve + twice the clearance between one end of valve and steam chest.

$c = .75\ i$.

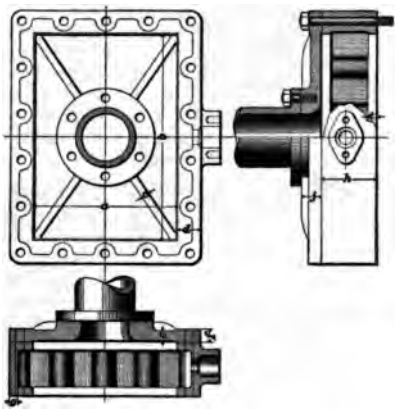


FIG. 2.

$d = 2.75\ o$, where o is the nominal diameter of the steam chest bolts, as in Fig. 1.

$e = .04\ \sqrt[4]{A'} + .125''$ for all areas above 100 sq. in. A' = area of steam chest, outside measurement, in square inches.

$f = 1.3\ e$. $g = .85\ i$.

h = height of valve + necessary clearance.

$t = .85\ i$. $= 2.5\ i$.

NOTE.—When the area of the steam chest cover is less than 100 square inches, its thickness e may be equal to i. When the area of the steam chest cover exceeds 600 square inches, the height of the ribs should be $3.5\ i$, and their number should be increased.

Fig. 3 shows a design for a steam chest cover when the steam pipe flange is on one side of the steam chest. Determine the thickness e by the same formula and rules as for the cover in Fig. 2. The other dimensions are found as follows:

$c = .75\ e.$ $\quad$ $j = 2.6\ c.$

$f = 1.3\ e.$ $\quad$ $r = 6\ e.$

p should never exceed the distance in inches given by the formula $\sqrt{\frac{40\ e_1^2}{Pg}}$, where e_1 is the numerator of the fraction expressing the thickness of the cover in sixteenths of an inch, and P_g = gauge boiler pressure.

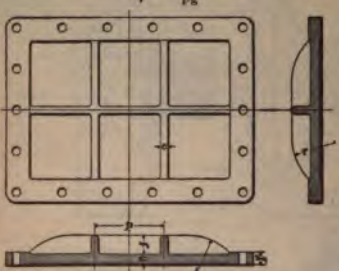


Fig. 3.

EXAMPLE. — Find the maximum pitch of the ribs for a cover $1\frac{15}{16}$" thick subjected to a steam pressure of 160 lbs. per square inch.

SOLUTION.—Substituting in the formula for p, we have

$$p = \sqrt{\frac{40 \times e_1^2}{Pg}} = \sqrt{\frac{40 \times 15^2}{160}} = 7.5 \text{ in.}$$

Fig. 4 shows a Corliss engine cylinder which may be designed according to the following proportions:

D = diameter of cylinder.

$a = 1.21\ D + 2\ e + 1.25''$.
$b = .2\ D + 1.125''$.
$c = .048\ D$.
$c' = .079\ D$.
$d = .17\ D$.
$e = .0003\ P\ D + .375''$, if boiler pressure is above 100 lb.; otherwise, $e = .03D + .375''$.
$f = .82\ e$.
$g = .9\ e$.
$h = b + 2\ (c + g)$.
$h' = h$.
$i = 1.8\ e$.
$j = e$.
$k = 1.2\ e$.
$l = 1.7\ x + 2'' - 1.2\ e$, where x = diameter of piston rod.
$l' = .32\ D$, about.
$m = .25\ D$.

n = .32 D.
o = 1.25 e.
p = 1.3 e.
q = .25 D.
q′ = .32 D.
r = 1.2 e.
s = 1.5 e.
t = (see note).
u = e, take nearest standard size bolt.
v = 1.2 e, take nearest standard size bolt.
w = 1.7 x + 2.25″, where x = diameter of piston rod.
y = D
z = 1.5e.

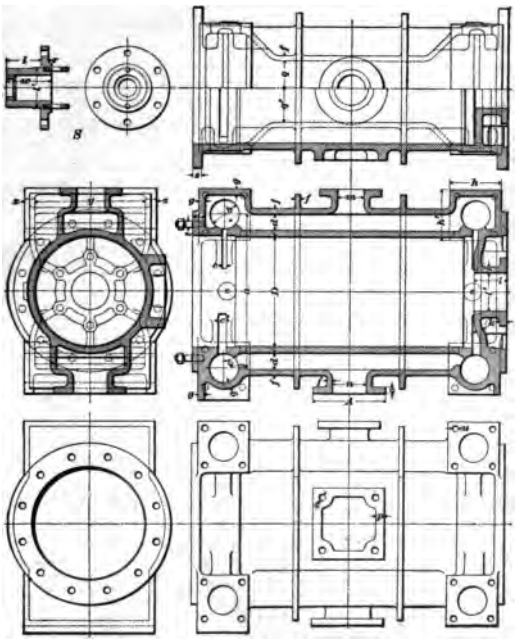


FIG. 4.

A is to be made according to proportions given on page 135.

Bolts to be made according to the same table.

NOTE.—The bolts for cylinder heads are to be calculated from the formula given for cylinder head bolts in connection with Fig. 1.

In this cylinder the stuffing box, *S*, is a separate piece that is to be bolted to the cylinder head.

Crank Shafts.

For high speed automatic short stroke engines, the following formula corresponds with good practice :

$$d = .44\ D + \tfrac{1}{2}'',$$

where d is the diameter of shaft and D = the diameter of cylinders.

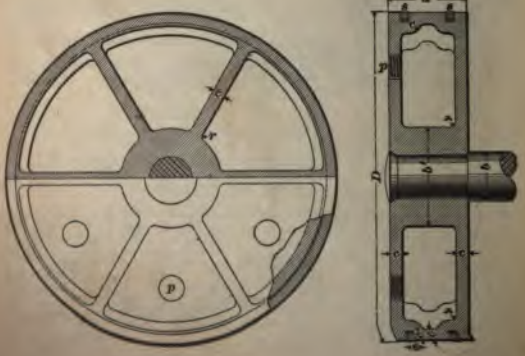
For the Corliss type, in which the stroke is equal to or greater than twice the diameter,

$$d = .34\ D + 2\tfrac{1}{2}'',$$

when D is equal to or greater than 16 inches. When D is less than 16 inches, $d = \tfrac{1}{2}\ D$.

The Piston.

A form of piston that is much used is shown in the following figure. It consists simply of a hollow circular disc of cast iron.



The packing rings s, s are made of cast iron, and are split and sprung into place. Their elasticity causes them to press against the cylinder walls and thus prevent the leakage of steam.

The following proportions will give dimensions suitable for this piston :

D = diameter of cylinder in inches.

$a = .2\ D + 1.5''$.	$e = .75\ c$.
b = diameter of piston rod.	$r = .5\ c$.
$b' = 2\ b$.	p = core plug.
$c = .18\sqrt{2\ D} - .1875''$.	Number of ribs $= .08\ (D^2 + 34)$.

The figure shows a *strap end* connecting rod. The straps c_1 and c_2 are fastened to the ends of the rods by means of the gibs a_1 and a_2 and the cotters b_1 and b_2. The cotters are held in place by the set screws s_1 and s_2. Small steel blocks shown between the ends of the set screws and the cotters, are used to prevent injury of the cotter by the set screws.

The rod, cotters, gibs and straps may be made of either wrought iron or steel. The crank pin brasses are shown babbitted, and wrist pin brasses without babbitt. The brasses are adjusted by means of the cotters, which draw the straps farther on to the rod when they are driven in.

The dimensions for the rod are given by the following proportions :

For wrist pin end,

D = diameter of cylinder.	$f = .26\ D + .5''$ for cylinders to 26" in diameter, and $f = .28\ D$ for cylinders above 26" in diameter.
$d = .2\ D$ = diameter of wrist pin.	$g = 1.3\ n$.
$n = .155\ D + .0625''$.	$h = \frac{.5\ x}{g - c}$.
$x = \frac{\pi}{4} n^2$ = a factor for use in finding proportions below.	$i = \frac{.32\ x}{h}$.
$a = .75\ d + .125''$.	$k = \frac{1.8\ d}{x}$.
$a' = .75\ d + .125''$.	
$b = \sqrt{2.5\ x}$.	
$c = .25\ b$.	
$e = .125\ d$.	

Fig. 3 shows a design for a steam chest cover when the steam pipe flange is on one side of the steam chest. Determine the thickness e by the same formula and rules as for the cover in Fig. 2. The other dimensions are found as follows:

$c = .75\ e.$ $\quad$ $j = 2.6\ c.$

$f = 1.3\ c.$ $\quad$ $r = 6\ e.$

p should never exceed the distance in inches given by the formula $\sqrt{\frac{40\ e_1^2}{pg}}$, where e_1 is the numerator of the fraction expressing the thickness of the cover in sixteenths of an inch, and p_g = gauge boiler pressure.

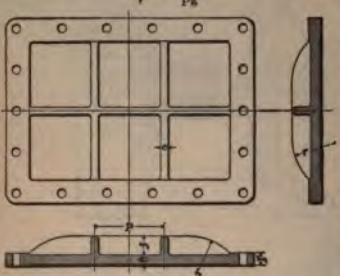


Fig. 3.

EXAMPLE. — Find the maximum pitch of the ribs for a cover $\frac{15}{16}$'' thick subjected to a steam pressure of 160 lbs. per square inch.

SOLUTION.—Substituting in the formula for p, we have

$$p = \sqrt{\frac{40 \times e_1^2}{pg}} = \sqrt{\frac{40 \times 15^2}{160}} = 7.5 \text{ in.}$$

Fig. 4 shows a Corliss engine cylinder which may be designed according to the following proportions:

D = diameter of cylinder.

$a = 1.21\ D + 2\ e + 1.25''$.
$b = .2\ D + 1.125''$.
$c = .048\ D$.
$c' = .079\ D$.
$d = .17\ D$.
$e = .0003\ P\ D + .375''$, if boiler pressure is above 100 lb.; otherwise, $e = .03D + .375''$.
$f = .82\ e$.
$g = .9\ e$.
$h = b + 2\ (c + g)$.
$h' = h$.
$i = 1.8\ e$.
$j = e$.
$k = 1.2\ e$.
$l = 1.7\ x + 2'' - 1.2\ e$, where x = diameter of piston rod.
$l' = .32\ D$, about.
$m = .25\ D$.

n = .32 D.
o = 1.25 e.
p = 1.3 e.
q = .25 D.
q' = .32 D.
r = 1.2 e.
s = 1.5 e.
t = (see note).
u = e, take nearest standard size bolt.
v = 1.2 e, take nearest standard size bolt.
w = 1.7 x + 2.25", where x = diameter of piston rod.
y = D
z = 1.5e.

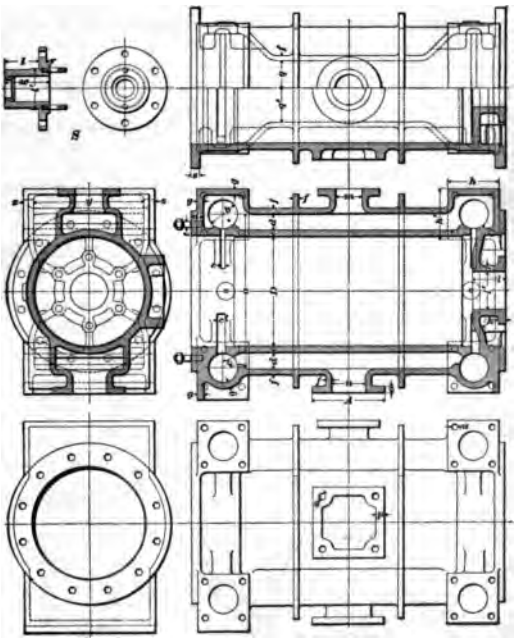
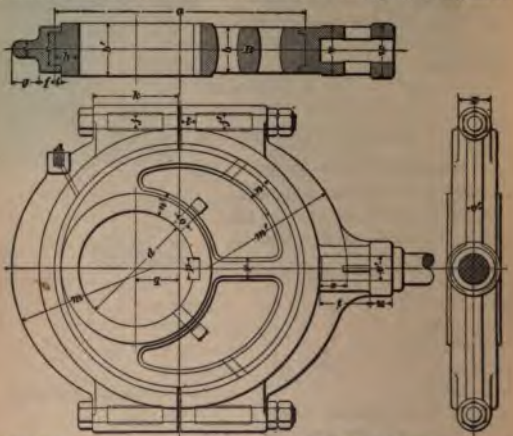


FIG. 4.

Eccentric and Strap.

The figure shows an eccentric sheave and strap, both made of cast iron. The eccentric sheave is cast solid, and must, there



fore, be slipped over the end of the shaft. The eccentric rod is held in a boss on the strap by means of a cotter.

For eccentrics used with valve stems $\frac{1}{16}$" in diameter or less the holes for bolts, *j*, are not to be cored.

A = boss for oil cup. B = cross-section of rib *r*.

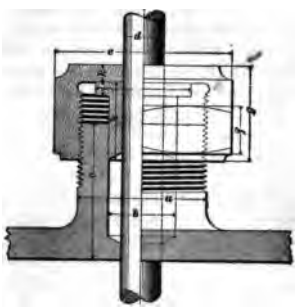
The proportions are as follows:

D = diameter of valve stem.	f = .7 D.
d = diameter of shaft.	g = 1.25 d.
a = d + 2 q + 2 h.	h = D + .125".
b = 2 D + .125".	i = .25 D + .0625".
b' = 2.25 D + .125".	j = area of bolt at root of thread = .38 D^2; use the nearest standard size bolt.
c = 1.5 D.	
e = .75 D.	
e' = .75 D.	

$j' = j + .1875.$
$k = 4\ D.$
$l = j.$
$m = \frac{d + 2q + 2h + f.}{2}$
$m' = m.$
$n = D + .125''.$
$n' = D + .125''.$
$o = .75\ j.$
$p = D.$
q = eccentricity.
$r = D.$
$s = 1.25\ D.$
$t = 2.25\ D + 1.25''.$
$u = D.$
$v = 2.25\ D.$
$v' = 1.125\ D.$
$w = 2.5\ D.$
$x = 2.25\ j.$

Stuffing Boxes.

The stuffing box of the form shown in the figure is generally used for small work, such as the spindles of valves, etc. The outside of the stuffing box is threaded to receive a hexagonal nut which fits over the gland. As the nut is screwed down, the gland is pressed downward and compresses the packing.



The proportions used are:

d = diameter of rod.
$a = 2.5\ d + .5''.$
$b = 1.5\ d + .125''.$
$c = 3 + d\ .25''.$
$e = 3.5\ d + .625''.$
$f = d + .125''.$
$g = 2\ d + .25''.$
$h = 1.5\ d + .25''.$
$i = .25\ d + .0625''.$
$k = .5\ d.$

This design may be used for rods up to 1¼″ diameter.

Make the number of threads per inch the same as for a bolt whose diameter is equal to the diameter of the rod.

Watt hour is a unit of work. It is used to indicate the expenditure of an electrical power of one watt for one hour.

EXAMPLE.—The resistance of a lighting circuit is 5 ohms and the electromotive force is 110 volts; (*a*) what is the amount of electrical power in watts required for this current? (*b*) What is the equivalent horsepower?

$$E = 110. \quad R = 5.$$

$$\frac{E^2}{R} = \frac{110^2}{5} = 2{,}420 \text{ watts.}$$

$$\frac{E^2}{R \times 746} = \frac{110^2}{5 \times 746} = 3.244 \text{ H. P.}$$

Conductivity is the name given to the reciprocal of the resistance of any conductor. There is no unit by which to express conductivity.

NOTE.—*The reciprocal of any number is unity divided by that number.*

Thus:—The reciprocal of 2 is $\frac{1}{2}$ or .5.

Rules for Direction of Current, etc.

To determine the direction of a current in a conductor by the aid of a compass:

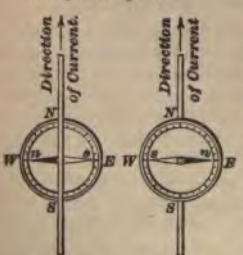


FIG. 1. FIG. 2.

RULE.—If the current flows from the south pole over the needle to the north, the north end of the needle will point towards the west, as in Fig. 1. If the compass is placed over the conductor so that the current will flow from the south under the needle to the north, the north end of the needle will point toward the east, as in Fig. 2.

To determine the polarity of an electro-magnet:

RULE.—In looking at the face of a pole (Fig. 3), if the current flows in the direction *a*, of the hands of a watch, it will be a *south* pole, and if in the opposite direction *b*, it will be a *north* pole.

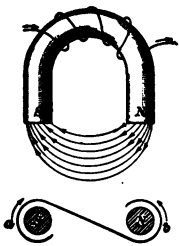


FIG. 3.

To determine the direction of an induced current in a conductor which is moving in a magnetic field:

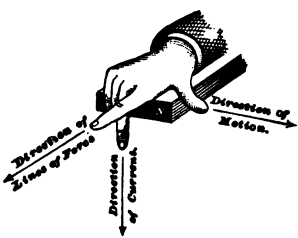


FIG. 4.

RULE.—Place thumb, forefinger and middle finger of *right hand*, each at a right angle to the other two, as shown in Fig. 4; if the *forefinger* shows direction of *lines of force* and the thumb the direction of *motion of conductor*, then the middle finger will show the direction of the induced current.

NOTE.—The above rule will give the polarity of a dynamo.

To determine the direction of motion of a conductor carrying a current when placed in a magnetic field:

FIG. 5.

RULE.—Place thumb, forefinger and middle finger of the *left hand*, each at a right angle to the other two, as shown in Fig. 5; if the forefinger shows the direction of the *lines of force* and the *middle finger* shows the direction of the current, then the thumb will show the direction of motion of the conductor.

NOTE.—The above rule will give the polarity of a motor.

Derived or Shunt Circuits.

A circuit divided into two or more branches, each branch transmitting part of the current, is said to be a *derived circuit*; the individual branches are in multiple-arc, or parallel with each other.

To find the joint resistance of a derived circuit:

As the conductivity of any conductor is equal to the reciprocal of its resistance, then the joint conductivity of two or more circuits in parallel is equal to the sum of the reciprocals of their separate resistances. The joint resistance of two or more circuits in parallel is equal to the reciprocal of their joint conductivity.

In a derived circuit of three branches, let r_1, r_2, and r_3 equal the resistance of each branch respectively. Their joint conductivity, or the sum of the reciprocals of their resistances, is—

$$\frac{1}{r_1} + \frac{1}{r_2} + \frac{1}{r_3}, \text{ or } \frac{r_2 r_3 + r_1 r_2 + r_1 r_2}{r_1 r_2 r_3}.$$

Their joint resistance is, therefore,

$$\frac{1}{\frac{r_2 r_3 + r_1 r_3 + r_1 r_2}{r_1 r_2 r_3}}, \quad \text{or} \quad \frac{r_1 r_2 r_3}{r_2 r_3 + r_1 r_3 + r_1 r_2}.$$

The joint resistance of a derived circuit with but two branches in parallel may be thus expressed:

$$\frac{\text{Product of their resistances}}{\text{Sum of their resistances}}.$$

EXAMPLE.—The resistances of two branches of a derived circuit are 20 and 30 ohms respectively. Find their joint resistance.

$$\frac{\text{Product of their resistances}}{\text{Sum of their resistances}} = \frac{600}{50} = 12 \text{ ohms.}$$

To find the strength of current in the separate branches of a derived circuit:

RULE.—A current is divided among the branches of a derived circuit in proportion to their conductivities—*i. e.*, to the reciprocal of their resistances.

EXAMPLE.—If the resistance of the two branches, A and B, of a derived circuit are 20 and 30 ohms respectively, and the total current in the main circuit is 60 amperes, what is the current in each? The conductivity of A is $\frac{1}{20}$ and of B $\frac{1}{30}$,

If C_1 represents the current in A, and
C_2 " " " " B,
then $C_1 : C_2 :: \frac{1}{20} : \frac{1}{30}$.

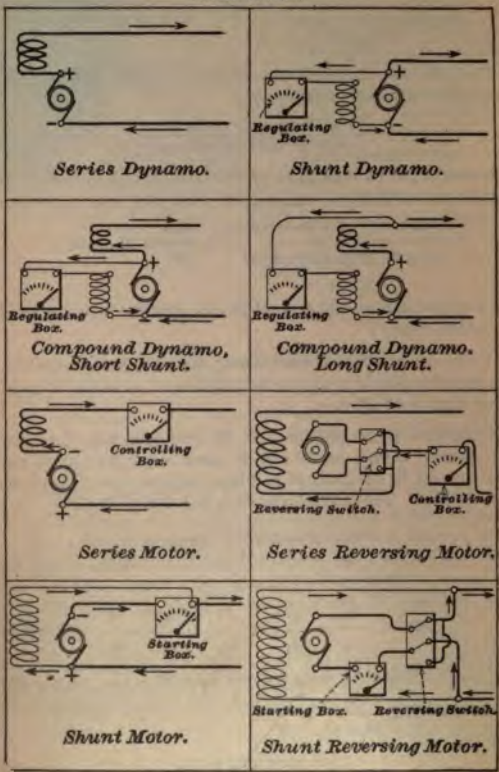
Hence, $\frac{C_1}{C_2} = \frac{\frac{1}{20}}{\frac{1}{30}}$, or $\frac{C_1}{C_2} = \frac{30}{20} = \frac{3}{2}$.

Now $C_1 + C_2 = 60$, or $C_2 = 60 - C_1$.

Substituting $\frac{C_1}{60 - C_1} = \frac{3}{2}$

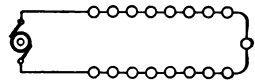
$C_1 = 36$ and $C_2 = 24$.

CONNECTIONS FOR DYNAMO-ELECTRIC MACHINES.

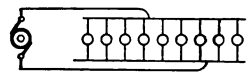


In the diagrams showing the connections of dynamo-electric machines, the heavy coils represent the series winding on the field-magnets through which the entire current of the machine passes; the lighter coils represent the shunt winding on the field-magnets through which only part of main current passes.

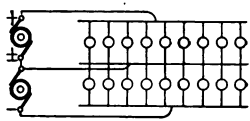
Lamps connected in series.



Lamps connected in multiple-arc or parallel.



Edison three-wire system.



Interior Wiring.

A *mil* is a unit of length used in measuring the diameters of wires, and is equal to .001 inch.

A *circular mil* is a unit of area used in measuring the cross-sections of wires, and is equal to $\frac{.7854}{10^6}$ sq. in.

The sectional area of a wire expressed in circular mils is equal to the square of its diameter in mils.

Let c. m = circular mils.
C = total current in amperes.
c = current in amperes to each lamp.
n = number of lamps in multiple.
v = volts lost in line.
r = resistance per foot of wire.
d = distance from dynamo to lamps.

The resistance of one foot of commercial copper wire, one mil in diameter, at a temperature of 75° F., is 10.8 ohms.

A 16 c. p. (candlepower) 110-volt lamp takes about .5 ampere; a 16 c. p. 55-volt lamp takes about 1 ampere.

All calculations for size of wire must be checked by comparing with a table of safe carrying capacity (see table on page 158), and the current value there given must not be exceeded.

To find the size of wire for 100-volt circuit with 16 c. p. lamps:

$$r = \frac{v}{n\,d},$$

$$\text{for large cables, c. m} = \frac{10.8\,n\,d}{v}.$$

EXAMPLE.—Find the size of wire necessary for a circuit supplying current to 50 100-volt 16 c. p. lamps, 300 feet from the dynamo, allowing a loss of 5 per cent. in line.

$$\text{Volts at dynamo} = \frac{110}{.95} = 115.8.$$

$$\text{Volts lost in line} = 115.8 - 110 = 5.8 = v.$$

$$\text{Then } r = \frac{v}{n\,d} = \frac{5.8}{50 \times 300} = .000386 \text{ ohm per foot,}$$

$$= .386 \text{ ohm per 1,000 feet.}$$

The nearest size of wire, as given in the table on page 158 is No. 6 B. & S., and its current capacity is 35 amperes, therefore it safe.

To find the size of wire for a 55-volt circuit with 16 c. p. lamps:

$$r = \frac{v}{2\,n\,d}$$

$$\text{for large cables, c. m} = \frac{21.6\,n\,d}{v}.$$

EXAMPLE.—What size of wire should be used for supplying current to 75 16 c. p. lamps on a 55-volt circuit, the distance from dynamo being 230 feet, and line loss, 4 volts?

$$r = \frac{v}{2\,n\,d} = \frac{4}{2 \times 75 \times 230} = .000116 \text{ ohm per foot,}$$

$$= .116 \text{ ohm per 1,000 feet.}$$

By reference to Table VII, the nearest wire is found to be No. 1 B. & S., and its carrying capacity is greater than the current, 75 amperes, which it is to conduct.

To find the size of wire for any circuit on a 2-wire system:

$$\text{In general, } r = \frac{v}{C \times 2d}$$

$$\text{or, c. m.} = \frac{10.8 \times 2d \times C}{v}.$$

EXAMPLE.—What wire should be used to carry 450 amperes a distance of 600 feet, the allowable drop being 6 per cent., and the E. M. F. at the end of the circuit 115 volts?

$$\text{Volts at dynamo} = \frac{115}{.94} = 122.3.$$

$$\text{Volts lost in line} = 7.3.$$

$$\text{Then, c. m.} = \frac{10.8 \times 2 \times 600 \times 450}{7.3} = 798{,}900.$$

Comparing this figure with the table on page 159 giving current capacity of cables, it will be seen that it is within the prescribed limits.

These formulas may be used for feeders, mains, branch mains, service mains, and inside wiring on continuous-current circuits, and for secondary wiring on alternating systems.

To find the size of wire for a 110-volt circuit, 3-wire system, 16 c. p. lamps:

$$r = \frac{4v}{nd} \quad \text{for each wire.}$$

$$\text{For large cables, c. m.} = \frac{2.7\,nd}{v} \quad \text{"}$$

In checking for carrying capacity, remember that the wire carries only one-half the current that would be used on a two-wire system, as the voltage between the outside conductors is double the voltage at the terminal of one lamp.

EXAMPLE.—What should be the size of the conductors for a three-wire system, when 132 100-volt, 16 c. p. lamps are installed at a distance of 210 feet from the source of supply, the loss being 4 volts?

$$r = \frac{4 \times 4}{132 \times 210} = .000577 \text{ ohm per foot,}$$

$$= .577 \text{ ohm per 1,000 feet.}$$

This would call for a wire between Nos. 7 and 8. The current will be $\frac{132 \times .5}{2} = 33$ amperes; but this is too much for the wire to carry, and No 6 B. & S. wire should be used, notwithstanding the somewhat less drop in volts which will result.

For continuous-current circuits, 5% less is usually allowed, with full current, from the dynamo to the lamps. For long distances a larger line loss may be allowed, if the dynamo is wound for that loss.

DIMENSIONS, WEIGHT AND RESISTANCE OF COPPER WIRE.

B. & S. Gauge.	Diameter in Mils (d). 1 mil = .001 in.	Area.	Weight and Length.		Resistance at 75° F. Ohms per 1,000 ft.	Current Amperes		B. & S. Gauge.
		Circular Mils. (d^2).	Lbs. per 1000 ft.	Feet per Pound.		Exposed.	Concealed.	
0000	460.000	211600.0	639.33	1.56	.049	300	175	0000
000	409.640	167805.0	507.01	1.97	.062	245	145	000
00	364.800	133079.0	402.09	2.49	.078	215	120	00
0	324.950	105592.0	319.04	3.13	.098	190	100	0
1	289.300	83694.0	252.88	3.95	.124	160	95	1
2	257.630	66373.0	200.54	4.99	.156	135	70	2
3	229.420	52634.0	159.03	6.29	.197	115	60	3
4	204.310	41742.0	126.12	7.93	.248	100	50	4
5	181.940	33102.0	100.01	10.00	.313	90	45	5
6	162.020	26250.0	79.32	12.61	.395	80	35	6
7	144.280	20817.0	62.90	15.90	.498	67	30	7
8	128.490	16509.0	49.88	20.05	.628	60	25	8
9	114.430	13094.0	39.56	25.28	.792			9
10	101.890	10381.0	31.37	31.88	.999	40	20	10
11	90.742	8234.1	24.88	40.20	1.260			11
12	80.808	6529.9	19.73	50.69	1.589	30	15	12
13	71.961	5178.4	15.65	63.91	2.003			13
14	64.084	4106 8	12.41	80.59	2.526	22	10	14
15	57.068	3256.7	9.83	101.65	3.186			15
16	50.820	2582.9	7.80	128.17	4.017	15	5	16
17	45.257	2048.2	6.19	161.59	5.066			17
18	40.303	1624.3	4.91	203.76	6.388	10		18
19	35.890	1252.4	3.89	257.42	8.055			19
20	31.961	1021.5	3.08	324.12	10.158	5		20

CARRYING CAPACITY OF CABLES.

Area Circular Mils.	*Open.*	*Concealed.*	*Area Circular Mils.*	*Open.*	*Concealed.*
200,000	299	200	1,200,000	1,147	715
300,000	405	272	1,300,000	1,217	756
400,000	503	336	1,400,000	1,287	796
500,000	595	393	1,500,000	1,356	835
600,000	682	445	1,600,000	1,423	873
700,000	765	494	1,700,000	1,489	910
800,000	846	541	1,800,000	1,554	946
900,000	924	586	1,900,000	1,618	981
1,000,000	1,000	630	2,000,000	1,681	1,015
1,100,000	1,075	673			

To find the size of wire on primary circuits for alternating system:

$$\text{c. m.} = \frac{10.8 \times 2\,d \times C^1}{v}$$

$$r = \frac{v}{C^1 \times 2\,d}.$$

C^1 = the total current in amperes on primary circuit and may be determined by dividing the total current on the secondary circuit, by the product of the ratio and efficiency of conversion.

The ratio is generally 20 to 1 on a 1,000-volt apparatus when using 52-volt lamps, and 10 to 1 when using 100- to 110-volt lamps.

The efficiency of conversion can be taken as 95% in ordinary transformers.

EXAMPLE.—If the loss is 5%, find the size of wire necessary on a 1000-volt primary circuit when the distance between the dynamo and transformer is 2000 ft. and the dynamo is supplying current for 500 16 c. p., 52-volt lamps.

$$\text{Volts at dynamo} = \frac{1000}{.95} = 1052, \text{ nearly.}$$

Volts lost in line = 52.

Assume the *lamp efficiency* to be 3.6 watts per c. p. Then, since the product of amperes and volts gives watts.

$$\text{Current to each lamp} = \frac{3.6 \times 16}{52} = 1.11 \text{ amperes.}$$

$$\text{Current on secondary} = 1.11 \times 500 = 555 \text{ amperes.}$$

$$\text{Total current on primary is } \frac{555}{.95 \times 20} = 29.21 \text{ amperes.}$$

$$\text{Therefore, c. m.} = \frac{10.8 \times 2\,d \times c^1}{v} = \frac{10.8 \times 4000 \times 29.21}{52} = 24267.$$

$$\text{And } r = \frac{v}{C^1 \times 2\,d} = \frac{52}{29.21 \times 4000} = .000445 \text{ ohm per ft., or .445 ohm per 1,000 ft.}$$

In the table on page 158, the nearest size of wire corresponding to this area and resistance is No. 6, B. & S. gauge.

For alternating systems under ordinary conditions, 5% loss at full load from dynamo to transformer on primary circuit is a maximum, although some dynamos are specially wound for 10% loss. A loss of from 1% to 2% may be allowed on secondary circuits from transformer to lamps.

Incandescent Lamp Data.

Let c = current in amperes to each lamp.

E = electromotive force in volts.

$$R = \frac{E}{c} = \text{resistance of lamp when hot.}$$

c. p. = candle power of lamp.

W. per c. p. = Watts per candle power (sometimes called *lamp efficiency*).

$$\text{W. per c. p.} = \frac{c \times E}{\text{c. p}}.$$

$$\text{The number of candles per electrical H. P.} = \frac{746}{\text{W. per c. p.}}$$

$$c = \frac{\text{W. per c. p.} \times \text{c. p.}}{E}.$$

As the commercial efficiency of good dynamos is about 90%, the calculations of candles per electrical H. P. must be multiplied by .90 to give the number of candles per mechanical H. P.

LAMP EFFICIENCIES.

3.1	watts	per	c.	p.,	or	12	lamps,	16	c. p.,	to 1 mechanical H. P.
3.6	"	"	"	"	10	"	"	"	1	" "
4.0	"	"	"	"	8	"	"	"	1	" "

NOTE.—Lamps of an efficiency of 3.1 watts per c. p. should not be used where the voltage averages, for any length of time, more than 2% high; lamps of 3.6 watts per c. p. should not be used where the voltage averages more than 4% high; and lamps of an efficiency of 4 watts per c. p. should be used where the regulation of the plant receives little or no attention. If these cautions are not followed, the life of the lamp will be greatly diminished.

Size of wire for arc-light circuits:

For ordinary distances, or small currents, use No. 8, B. & S. wire. For longer distances, or large currents, use No. 6, B. & S. wire.

Motor Circuits.

To find the size of wire on stationary motor circuits:

Let c. m. = circular mils.
e = E. M. F. of motor in volts.
v = loss of volts in line.
d = distance from generator to motor in feet.
k = efficiency of motor.

10.8 ohms is the resistance of one foot of commercial copper wire one mil in diameter.

$$\text{c. m.} = \frac{\text{H.P. of motor} \times 746 \times 2\,d \times 10.8}{e\,v\,k}.$$

APPROXIMATE MOTOR EFFICIENCY.

¾	to 1½	H. P. inclusive	= 75% efficiency.
3	to 5	" "	= 80% "
7½	to 10	" "	= 85% "
15 H. P. and upwards			= 90% "

Under ordinary circumstances, 10% loss from generator to motor is a maximum on stationary motor circuits.

EXAMPLE.—What is the size of wire necessary for a circuit on which a 10 H. P. 500-volt motor is running, when the distance between the motor and generator is 2,000 feet and the loss is 5%?

$$\text{Volts at generator } \frac{500}{.95} = 526, \text{ nearly.}$$

Volts lost in line $526 - 500 = 26$.

In the table above, the approximate efficiency of a 10-H. P. motor is given 85%.

$$\text{c. m.} = \frac{10 \times 746 \times 4000 \times 10.8}{500 \times 26 \times .85} = 29165.$$

In the table on page 158, the nearest size of wire corresponding to this area is No. 6, B. & S. gauge.

The approximate weight and resistance per mile of round bare wire when d is the diameter in mils are, for copper wire, $\frac{d^2}{62.5}$ lbs. and $\frac{56970}{d^2}$ ohms; for iron wire, $\frac{d^2}{72}$ lbs. and $\frac{380000}{d^2}$ ohms.

Copper wire is approximately 1¼ times the weight of an iron wire of the same diameter.

In determining the size of wire to be used for inside work, after finding the c. m., always refer to the table on page 158, and see that the wire obtained by the formula is sufficiently large to carry the current; if not, use larger wire regardless of per cent. loss. *For pole-line construction, never use wire smaller than No. 8, B. & S. gauge.*

EQUIVALENT SECTIONAL AREA OF WIRES, B. & S. GAUGE.

Gauge No.	No. of Wires. — Gauge No.	No. of Wires. — Gauge No.	No. of Wires. — Gauge No.	No. of Wires. — Gauge No.	No. of Wires. — Gauge No.	No. of Wires. — Gauge No.	Gauge No. — Gauge No.
0000	2– 0	4– 3	8– 6	16– 9	32–12	64–15	
000	2– 1	4– 4	8– 7	16–10	32–13	64–16	
00	2– 2	4– 5	8– 8	16–11	32–14	64–17	1 and 3
0	2– 3	4– 6	8– 9	16–12	32–15	64–18	2 " 4
1	2– 4	4– 7	8–10	16–13	32–16		3 " 5
2	2– 5	4– 8	8–11	16–14	32–17		4 " 6
3	2– 6	4– 9	8–12	16–15	32–18		5 " 7
4	2– 7	4–10	8–13	16–16			6 " 8
5	2– 8	4–11	8–14	16–17			7 " 9
6	2– 9	4–12	8–15	16–18			8 " 10
7	2–10	4–13	8–16				9 " 11
8	2–11	4–14	8–17				10 " 12
9	2–12	4–15	8–18				11 " 13
10	2–13	4–16					12 " 14
11	2–14	4–17					13 " 15
12	2–15	4–18					14 " 16
13	2–16						15 " 17
14	2–17						16 " 18
15	2–18						

The above table indicates the number of smaller wires required to give a sectional area equal to one larger size wire ; the figures between the horizontal lines corresponding to each other. For example : It requires 2 wires, No. 0, or 4 wires, No. 3, etc., to give a sectional area equal to 1 wire, No. 0000. Again : it requires 2 wires, No. 13, or 4 wires, No. 16 ; or, 2 wires, 1 No. 12 plus 1 No. 14, to give a sectional area equal to 1 No. 10.

COMPARATIVE SIZES OF WIRES, B. & S. AND BIRMINGHAM GAUGES.

Diameter, Inches.	*B. & S.*	*Birmingham.*
.460	0000	
.454		0000
.425		000
.4096	000	
.380		00
.3648	00	
.340		0
.3249	0	
.3000		1
.2893	1	
.284		2
.259		3
.2576	2	
.238		4
.2294	3	
.22		5
.2043	4	
.203		6
.1819	5	
.18		7
.165		8
.162	6	
.148		9
.1443	7	
.134		10
.1285	8	
.12		11
.1144	9	
.109		12
.1019	10	
.095		13
.0907	11	
.083		14

COMPARATIVE SIZES OF WIRES, B. & S. AND BIRMINGHAM GAUGES.—*Continued.*

Diameter, Inches.	*B. & S.*	*Birmingham.*
.0808	12	
.072	13	15
.065		16
.0641	14	
.058		17
.0571	15	
.0508	16	
.049		18
.0453	17	
.042		19
.0403	18	
.0359	19	

NOTE.—B. & S. gauge is generally used in America.

WEIGHT OF UNDERWRITERS' LINE WIRE, INSULATED.

No. B. & S.	*Pounds per 1,000 Feet.*	*Feet per Pound.*
0000	800	1.25
000	666	1.50
00	500	2.00
0	363	2.75
1	313	3.20
2	250	4.00
3	200	5.00
4	144	6.9
5	125	8.0
6	105	9.5
7	87	11.5
8	69	14.5
10	50	20.0
12	31	32.0
14	22	45.0
16	14	70.0
18	11	90.0

RESISTANCE OF PURE COPPER WIRE.

No. B. & S.	Resistance at 75° F.			
	R. Ohms per 1,000 Feet.	*Ohms per Mile*	*Feet per Ohm.*	*Ohms per Pound*
4-0	.04904	.25891	20392.9	.00007653
3-0	.06184	.32649	16172.1	.00012169
00	.07797	.41168	12825.4	.00019438
0	.09827	.51885	10176.4	.00030734
1	.12398	.65460	8066.0	.00048920
2	.15633	.82543	6396.7	.00077784
3	.19714	1.04000	5072.5	.0012370
4	.24858	1.31248	4022.9	.0019666
5	.31346	1.65507	3190.2	.0031273
6	.39528	2.08706	2529.9	.0049728
7	.49845	2.63184	2006.2	.0079078
8	.62849	3.31843	1591.1	.0125719
9	.79242	4.18400	1262.0	.0199853
10	.99948	5.27726	1000.5	.0317046
11	1.2602	6.65357	793.56	.0505413
12	1.5890	8.39001	629.32	.0803641
13	2.0037	10.5798	499.06	.127788
14	2.5266	13.3405	395.79	.203180
15	3.1860	16.8223	313.87	.323079
16	4.0176	21.2130	248.90	.513737
17	5.0660	26.7485	197.39	.816839
18	6.3880	33.7285	156.54	1.298764
19	8.0555	42.5329	124.14	2.065312
20	10.1584	53.6362	98.44	3.284374

Dynamo Design.

The fundamental principle of dynamo design is obtained in the formula $E = \frac{N C n}{10^8 \times 60}$, in which E is the electromotive force in volts given by the dynamo; N is the number of lines of force used to magnetize the armature; C is the number of conductors in a bi-polar machine, measured all round the outside of the armature (whether in one or more layers), or in a multipolar machine, as measured from a point opposite one north pole to a corresponding point opposite the next succeeding north pole; and n is the number of revolutions per minute of the armature.

EXAMPLE.—A two-pole dynamo has 2,000,000 lines of force passing from the north pole through the armature to the south pole; there are 200 conductors on the surface of the armature, and the speed is 1,500 revs. per min. The electromotive force generated will then be

$$E = \frac{2{,}000{,}000 \times 200 \times 1{,}500}{100{,}000{,}000 \times 60} = 100 \text{ volts.}$$

If a four-pole dynamo were used, having a four-circuit armature and four sets of brushes, with 1,000,000 lines of force passing through any one pole-piece, then the total number would be 2,000,000, because the same lines of force pass into a south pole that emerge from a north pole. With the same armature as above, the number of conductors to be counted is only 100, as taken from one north pole to the next, and the electromotive force =

$$E = \frac{2{,}000{,}000 \times 100 \times 1{,}500}{100{,}000{,}000 \times 60} = 50 \text{ volts.}$$

In predetermining the number of lines of force required in a specific case, the above formula may be reversed, and we have $N = \frac{E \times 10^8 \times 60}{C n}$. These lines of force have a circuit to *traverse composed of* three different paths. One of these is

through the field-magnet and yoke, *M* Fig. 1 ; next, through a double air gap *G*, and, lastly, through the armature core *A*. A given density of lines of force may not be exceeded, this limit being for ordinary cast iron about 50,000 lines per square inch ; for wrought iron forgings or cast steel, about 90,000, and for soft sheet iron, 110,000. The ratio of magnetization to magnetizing force is called the *permeability*. The permeability of air is very low, the intensity of magnetization being a direct measure of the magnetizing force required ; therefore, the air-gap is usually made short.

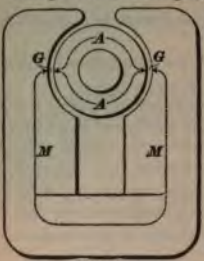


FIG. 1.

In order to drive the lines of force through the magnetic circuit, magnetizing coils are wound on the cores at *M*, *M*. A certain number of ampere-turns will be required, depending on the density of the lines of force and the permeability of the different portions of the circuit. The number of turns may be found by taking a convenient current value, and dividing the ampere-turns by this. Reference to a wire table will then determine whether the resistance of the wire will be such that the terminal E. M. F. of the machine will supply the proper current. A margin should be allowed for regulating, and for the increase in resistance due to rise in temperature, which is about .04 per cent. for every degree Centigrade, or 0.222 per cent. for every degree Fahrenheit above 75° F.

In the saturation curves of Fig. 2 are represented graphically the different values of the induction (**B**) in lines of force per square inch, corresponding to the magnetizing force expressed in ampere-turns per inch of length of circuit. Thus, to send 70,000 lines of force through a cast steel core one square inch in cross-sectional area, would require about 30 ampere-turns for every inch in length of core. The 30 ampere-turns might be obtained by using a coil of 30 turns carrying 1 ampere, or 300 turns of $\frac{1}{10}$ ampere, etc. The number of lines of force N for any particular case being known, and also the allowable density

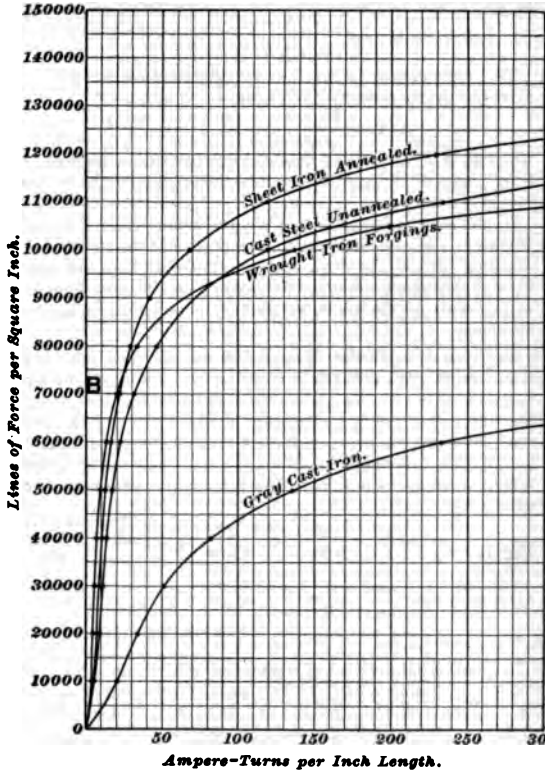

150000
140000
130000
120000
110000
100000
90000
80000
70000
60000
50000
40000
30000
20000
10000
0
B
Sheet Iron Annealed.
Cast Steel Unannealed.
Wrought Iron Forgings.
Gray Cast-Iron.
50
100
150
200
250
300
Lines of Force per Square Inch.
Ampere-Turns per Inch Length.

FIG. 2.

B which will vary somewhat with different samples of iron, the cross-sectional area $A = \frac{N}{B}$.

The ampere-turns to be added to the magnetizing coils to overcome the resistance of the air-gap is

$$A.\ T. = \frac{H \times l}{3.192},$$

where **H** is the number of lines of force per square inch, and *l* is the length of air-gap (the two sides added together) in inches —usually a fraction.

It is necessary, in calculating the ampere-turns for the field circuit, to allow for leakage of lines of force through the surrounding air, as the total number generated does not pass through the armature core. This leakage may amount to 30 or 40 per cent. of the whole, but is much less in well designed machines.

EXAMPLE.—A bi-polar dynamo has magnet cores having a mean length, with pole-pieces of 10 in. each; the yoke of the magnet is 13 in.; air-gap, $\frac{3}{16}$ in. each side; armature core, 10 in. The magnetic density in the core is 85,000; air-gap, 46,000; yoke, 65,000; armature core, 90,000 lines of force per square inch. If the fields are wrought iron forgings, and the armature is built up of soft sheet iron, then the ampere-turns necessary will be:

	Length.	**B**	*A. T. per in.*	*Ampere-turns.*
Magnet cores.....	20 in.	85,000	44	880
Yoke.....	13 "	65,000	16	208
Armature.....	10 "	90,000	40	400
Air-gap.....	$\frac{3}{8}$ "	46,000	..	5425
			Total ampere-turns	6,913

In determining the size of wire to be used in the armature winding, a certain density of current may be assumed as the limit. This is usually expressed in circular mils or thousandths of an inch per ampere. For most purposes of design, a density of 600 circular mils per ampere may be allowed. In estimating *the current passing* through the armature, it must be remembered *that the* current of the outside circuit divides on reaching

the armature, and passes through it along two paths in parallel with each other.

Faults of Dynamos.

Reversal of Field.—Run the machine up to speed and hold a small compass near each pole-piece in succession. Their polarity should alternate all the way round.

Failure to build up.—This is probably due to reversal of shunt connections. Rock the brushes around until any one set occupies a position formerly occupied by the next set. If this should remedy the trouble, and such position is inconvenient, move them back and reverse connections of shunt windings. If the failure of machine is due to want of residual magnetism, send a current from some external source through the fields. If it is due to a broken circuit, each coil may be tested separately with a battery and galvanometer or low-reading Weston voltmeter. Failure to generate may be due to the brushes being out of the neutral plane, which may be tested by moving them into different positions.

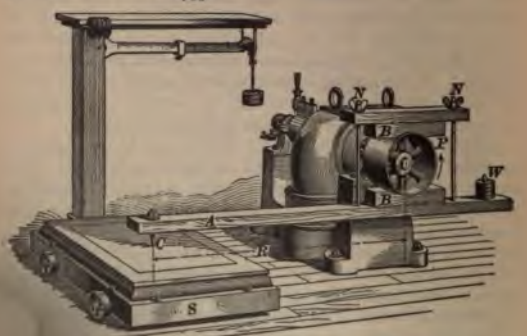
Heating.—This may be caused by a short-circuited armature coil. Allow the machine to cool, then run for a few minutes with no load, and stop. The defective coil will be found to be much hotter than the rest. It should be marked, and the armature taken out, when the coil may be rewound or otherwise repaired. If the heating is even, the load may be excessive and should be reduced, or the effect may be due to eddy currents in core, which is a question of design in the first instance.

Sparking at Commutator.—If this be due to overload, the sparking cannot be cured except by reducing the load. The trouble may be due to improper position of brushes. Move the rocker-arm to one side or the other to determine this. If copper brushes (tangential) are used, they may be unevenly spaced round the commutator; each set of brushes should have the same relative position with regard to the respective pole tips. Sparking may be caused by an uneven commutator, in which case it should be smoothed with sand paper (never emery), or turned down in the lathe. A broken connection at the commutator leads will produce flashing at each revolution, and one of the bars will show a burn extending nearly across it. The loose wire should be secured, or if broken, the commutator bars may

be connected together with a piece of wire or a drop of solder as a temporary repair. As soon as possible a new coil should be put in. Sparking may also occur, in a multipolar machine, from the wearing away of the bearings, which produces eccentricity of the armature with respect to field, and consequent unequal magnetic induction at different points. A slight sparking at the brushes is not detrimental.

Output and Efficiency of Motors.

A dynamo, when supplied with current from an external source, becomes a motor, turning the electrical energy into mechanical energy. The ratio between these two quantities, that is, between the input and output, determines the efficiency of the motor. The input may be found by measuring the current C with an ammeter, and the voltage E with a voltmeter, their product giving the power supplied in watts, W = C. E. This quantity divided by 746 gives the electrical horsepower, or E. H. P. $= \frac{W}{746}$.



he output is measured by means of a Prony brake, see re. *The motor* pulley *P* is clamped between two blocks of *B, B,* their pressure being regulated by the thumb screws

N, N, on the long bolts which hold them together. The lower block is extended to form an arm A of convenient length, and furnished with a sharp lag screw C at the end. The lag screw presses on the platform of a set of scales S whereby its pressure may be determined. A counterbalance at W neutralizes the weight of the arm. When the pulley is revolved in the direction shown, the pressure on the scale will indicate the torque or twisting power developed, which is expressed as the product of the pressure on the scale into the distance between the center of pulley and the point of the screw. If the length of arm $R = 2$ feet, and the pressure is 50 pounds, the torque $T = 100$ foot-pounds. The horsepower may be determined by the following formula:

$$\text{H. P.} = \frac{2 \times 3.1416 \, T \, S}{33{,}000}$$

in which S is the speed of motor in revolutions per minute.

EXAMPLE.—A motor takes a current of 40 amperes at 110 volts. The torque, when the motor is running at 1,000 revs. per min., is 27 foot-pounds. What is the efficiency?

$$\text{The input} = \frac{40 \times 110}{746} = 5.9 \text{ E. H. P.}$$

$$\text{The output} = \frac{2 \times 3.1416 \times 27 \times 1{,}000}{33{,}000} = 5.14 \text{ H. P.}$$

$$\text{The efficiency is then } \frac{5.14}{5.9} = 87.1 \text{ per cent.}$$

Applications of Electric Motors.

The same varieties of field and armature connections are used for motors as for dynamos, namely, series, shunt and compound, and each type has distinguishing characteristics. The series motor is especially suitable for use in cases where a very high starting torque is required in order to obtain rapid acceleration under load, as for instance, in street railway work. Torque may be defined as the reaction of the current in the armature or moving part against the magnetic lines of force in the field magnets or stationary part. Strength of field is obtained by the current circulating through the magnet coils; consequently, the torque in a series motor will be a maximum when the current passing through is a maximum, as the same amount flows through armature and field. The opposition to the flow of cur-

rent is the resistance of the circuit and the counter E. M. F. of the armature. When the current is applied, its value is determinable by Ohm's Law for the first moment, supposing self-induction to be eliminated. The resistance of a series motor is usually so low that an additional resistance must be used at starting in order to prevent an excessive flow of current: but, as soon as the armature commences to revolve, the counter E. M. F. opposes and cuts down the current and, consequently, the torque. The speed will continue to increase and the torque to decrease until the mechanical resistance to rotation balances the torque. If the motor is running light, the speed will rise continually, the counter E. M. F. will also increase and cut down the current, and the consequent reduction of field strength will require a still higher speed in order to develop the necessary counter E. M. F., the final result being, probably, the bursting of the armature. The speed of a series motor under a constant load may be regulated by the somewhat wasteful method of introducing a resistance in series to reduce the speed, and by cutting out or shunting part of the field coils, to increase it. When two motors are used, they may be put in series at starting and connected in parallel for higher speeds. The series motor is well adapted for electric cranes, because it will automatically regulate its speed to the weight to be raised, exerting a very powerful torque at low speed for a heavy load.

The shunt motor will give a nearly constant speed for any variation in load, as long as the potential of current supply (the applied E. M. F.) is constant. This condition produces a constant field, as the shunt winding is directly across the main leads, and the speed of the motor will then be such that the difference between the E. M. F. of supply and the counter E. M. F. divided by the resistance of the armature will be equal to the current passing through the armature. A change in the current will then produce but a relatively small change in the required counter E. M. F. of the motor, and the speed will only vary to that extent. As the load is put on, the motor tends to slow down; but this, by decreasing the counter E. M. F., allows *more current* to flow, thereby producing more torque to overcome the added mechanical resistance. Change of speed may be

produced by varying the strength of the magnetic field, the weaker the field the higher the speed. If the load is constant, the torque will be decreased, but if the load be correspondingly increased, the torque will remain nearly constant. Considerable weakening of the field is inadvisable, as it will cause destructive sparking at the commutator. The theoretically perfect method of speed regulation for a shunt motor is to provide a constant and independent field, and effect change of speed by varying the applied E. M. F. at the armature terminals without insertion of extra resistance. In this case the torque will always be proportional to the load, and the efficiency will be constant and independent of speed and torque. In the operation of such a system certain complications are introduced, inasmuch as it is necessary to install in connection with each motor a special dynamo with variable field, and this condition may therefore constitute a serious objection when the first cost of the plant is required to be low.

A differential compound winding may be used when a more nearly constant speed is wanted. The series turns on the field-magnets are so connected as to oppose the shunt turns, and when an increase of load tends to cut down the speed, the additional current through the series turns weakens the field slightly, so that the same speed as before is required to generate the lower counter E. M. F.

Shunt motors are especially useful for machine tools, which require a constant speed irrespective of load, and may also be used on printing presses and similar machines where the load is more nearly uniform. When a variation in speed with load is immaterial, a cumulative compound winding may be employed, in which the series turns act with the shunt, thereby increasing the torque at starting, and affording some of the characteristics of both the shunt and series windings.

Batteries.

The simple primary battery consists of two elements, the *anode*, which is usually zinc, and the *cathode*, which may be carbon, both immersed in an exciting liquid called the *electrolyte*. The chemical action incident to the generation of current dissolves the zinc and liberates free hydrogen at the cathode, which adheres to the surface and reduces the E. M. F. of the battery. To overcome this effect, called polarization, a depolarizor is used which will take up the hydrogen as it is formed.

Depolarizors may be solid or liquid. When solid, the material is usually packed round the cathode, as in the case of the Leclanche cell; when the depolarizor is liquid, it may be prevented from mixing with the electrolyte by a porous partition, or, if their specific resistances differ considerably, they will remain separated one over the other in the jar. The following table gives the elements and depolarizor for different cells, with the E. M. F. in volts:

Name.	Anode.	Electrolyte.	Cathode.	Depolarizor.	E. M. F.	Remarks.
Volta	Zinc	Sulphuric Acid (dilute)	Copper	None	.9	Polarizes rapidly.
Law	Zinc	Sulphuric Acid (dilute)	Carbon	None	1.35	
.....	Zinc	Sodium Chloride (common salt)	Carbon	None	1.08	
Grenet	Zinc	Sulphuric Acid 4 Potassium Bichromate 3 Water 18	Carbon	Electrolyte	1.9 to 2	For large currents.
Pabst	Wrought Iron	Ferric Chloride	Carbon	.....	.78	Non-polarizing electrolyte.
Bunson	Zinc	Sulphuric Acid (dilute)	Carbon	Nitric Acid	1.89	Cathode and depolarizor in porous cup.
Fuller	Amalgamated Zinc in Mercury	Sulphuric Acid (very dilute) or Water	Carbon	Electropoion Fluid diluted one-half	2.14	Anode in porous cup.
Partz	Zinc	Sodium Chloride or Magnesium Sulphate	Carbon	Bichromate solution (sulpho-chromic salt)	1.9 to 2	Gravity cell. Resistance with Sodium Chloride .5 ohm; with Magnesium Sulphate 1 ohm.

Name.	*Anode.*	*Electrolyte.*	*Cathode.*	*Depolarizor.*	*E. M. F.*	*Remarks.*
D'Arsonval.....	Zinc	Caustic Soda.....	Carbon...	Ferric Chloride.....	2.7	Porous cup used; pores become filled with ferric hydrate, an insoluble conductor.
Daniell...	Zinc	Zinc Sulphate.....	Copper...	Copper Sulphate with Copper Sulphate crystals....	1.07	Cathode and depolarizor in porous cup.
Gravity Daniell.	Zinc	Zinc Sulphate sp. gr. 1.10.....	Copper...	Copper Sulphate with Copper Sulphate crystals....	1.07	For closed circuit work only; resistance .5 ohm.
Leclanche	Zinc	Ammonium chloride (saturated).....	Carbon...	Peroxide of Manganese.....	1.48	Carbon and depolarizor in porous cup. Resistance 4 ohms.
Lalande and Chaperon	Zinc	Caustic Potash.....	Iron or Copper.	Cupric Oxide.....	.7	Surface of electrolyte covered with layer of oil.
Edison-Lalande.	Zinc	Caustic Potash.....	Molded plates of cupric oxide and magnesic chloride held in copper frames.....		.7	Surface of electrolyte covered with layer of oil. Resistance .07 ohm.
Chloride of Mercury cell	Zinc	Sal Ammoniac (Ammonium Chloride)	Carbon...	Paste of Mercurous Chloride	1.45	Cathode and depolarizor in porous cups. For small currents.

Name.	*Anode.*	*Electrolyte.*	*Cathode.*	*Depolarizor.*	*E. M. F.*	*Remarks.*
Volta....	Zinc.....	Sulphuric Acid (dilute).....	Copper...	None.....	.9	Polarizes rapidly.
Law.....	Zinc.....	Sulphuric Acid (dilute).....	Carbon...	None.....	1.35	
.....	Zinc.....	Sodium Chloride (common salt)....	Carbon...	None.....	1.08	
Grenet....	Zinc...	Sulphuric Acid 4 Potassium Bichromate 3 Water 18	Carbon...	Electrolyte.....	1.9 to 2	For large currents.
Pabst....	Wrought Iron....	Ferric Chloride.....	Carbon...	.....	.78	Non-polarizing electrolyte.
Bunson..	Zinc.....	Sulphuric Acid (dilute).....	Carbon...	Nitric Acid.....	1.89	Cathode and depolarizor in porous cup.
Fuller..	Amalgamated Zinc in Mercury	Sulphuric Acid (very dilute) or Water.....	Carbon...	Electropoion Fluid diluted one-half..	2.14	Anode in porous cup.
Partz.....	Zinc.....	Sodium Chloride or Magnesium Sulphate.....	Carbon...	Bichromate solution (sulpho-chromic salt).....	1.9 to 2	Gravity cell. Resistance with Sodium Chloride .5 ohm; with Magnesium Sulphate 1 ohm.

Name.	*Anode.*	*Electrolyte.*	*Cathode.*	*Depolarizor.*	*E. M. F.*	*Remarks.*
D'Arsonval.....	Zinc	Caustic Soda.....	Carbon...	Ferric Chloride.....	2.7	Porous cup used; pores become filled with ferric hydrate, an insoluble conductor.
Daniell...	Zinc	Zinc Sulphate.....	Copper...	Copper Sulphate with Copper Sulphate crystals....	1.07	Cathode and depolarizor in porous cup.
Gravity Daniell.	Zinc	Zinc Sulphate sp. gr. 1.10.....	Copper...	Copper Sulphate with Copper Sulphate crystals....	1.07	For closed circuit work only; resistance .5 ohm.
Leclanche	Zinc	Ammonium chloride (saturated).....	Carbon...	Peroxide of Manganese.....	1.48	Carbon and depolarizor in porous cup. Resistance 4 ohms.
Lalande and Chaperon	Zinc	Caustic Potash.....	Iron or Copper.	Cupric Oxide.....	.7	Surface of electrolyte covered with layer of oil.
Edison-Lalande.	Zinc	Caustic Potash.....	Molded plates of cupric oxide and magnesic chloride held in copper frames.....		.7	Surface of electrolyte covered with layer of oil. Resistance .07 ohm.
Chloride of Mercury cell	Zinc	Sal Ammoniac (Ammonium Chloride)	Carbon...	Paste of Mercurous Chloride	1.45	Cathode and depolarizor in porous cups. For small currents.

M. F. of the dynamo d is a little greater than that of the battery, so that it will charge the battery when there is no external load.

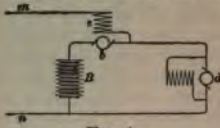


FIG. 1.

When all the lights are turned on, the booster field will be fully energized, and the E. M. F. of the booster will be added to that of the battery, thereby causing the battery to discharge and assist the dynamo. At a medium load, the battery will be neutral, neither taking current nor discharging, while the dynamo is running at full load. Any increase in load is then taken by the battery.

In electric railway plants the dynamos are usually over-compounded, giving a higher E. M. F. at the brushes at full load than at light load, and in this case a differential winding is employed, as shown in Fig. 2, causing the booster to work both

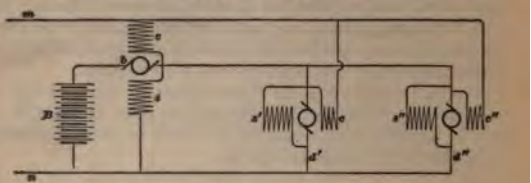


FIG. 2.

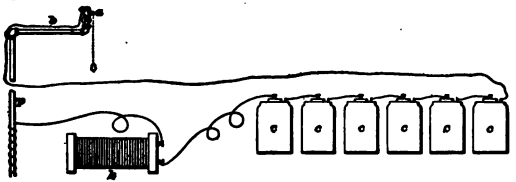
ways. On light loads it will assist the dynamos d' and d'' to charge the battery, raising the E. M. F. to the required value, but on heavy loads the series winding c will overpower the shunt s and the battery will discharge into the outer circuit. The shunt field must be regulated so that the total charging and discharging in a given time will balance each other, as the battery will otherwise tend either to overcharge or undercharge. *Strengthening* the shunt field will cause the batteries to charge, *and weakening* it will make them discharge at a lower value of *the external load* than before.

Electro Deposition.

For electro-deposition of metals, low resistance primary batteries giving from 2 to 10 volts may be used when the work is on a small scale. For larger work, accumulators may be employed, or the current may be taken directly from a low voltage dynamo. The electro-plating bath consists of a solution which has little or no chemical action on the objects to be plated, and which are suspended in it and electrically connected to the negative pole of the battery. The anode is a plate of the metal which it is desired to transfer; it is also submerged in the solution and connected through a resistance, if necessary, to the positive pole of the battery. For deposition of copper, the bath is made by taking four parts saturated solution of sulphate of copper mixed with one part of water containing one-tenth its volume of sulphuric acid. The current used must not exceed 18 amperes per square foot of surface of cathode. For nickel, use the double sulphate of nickel and ammonia, specific gravity, 1.03; the current density must be low, and the solution should be neutral or slightly alkaline, as an acid bath will cause the nickel to peel off. For silver, the bath is a solution of cyanide of silver dissolved in cyanide of potassium. For gold, use cyanide of gold dissolved in cyanide of potassium. This solution is kept at 150° F. while in use.

Electric Gas Lighting.

The arrangement of the apparatus required for electric gas lighting is shown in the figure. A battery of about 6 Leclanche



cells *c*, *c*, etc., joined up in series, is connected to one terminal of a spark coil *k*, the other terminal of which is soldered to a *gas pipe p*. The wire from the free end of the battery is carried

up through the house, and branches are run to the burners as at *b*, wherever needed. The insulation of this wire must be very thorough, special precautions being taken when it is carried through or along the fixtures. The burners are provided with a chain *a* attached to a movable contact spring which is drawn past the burner, producing a spark of sufficient intensity to ignite the gas if it is previously turned on.

In multiple gas lighting, a fine wire is run from one burner to another of the group as on a chandelier, leaving a small air-gap at each one, and a current of very high tension is used, generated by a small frictional machine, causing a spark at each burner. The last contact in a series of burners is connected to the gas-pipe.

The Wheatstone Bridge.

A diagrammatic sketch of the Wheatstone Bridge is shown in Fig. 1. This instrument is widely used for the determination of unknown resistances, and consists of such an arrangement of three circuits, *M*, *N*, *P*, of variable resistance, that the value of a fourth may be found from their relation. This unknown resistance is connected between the points *b* and *c*, and the battery *B* between *a* and *b*. The variable resistances are then so adjusted that there shall be no difference of potential between *c* and *d*, which form the terminals of the galvanometer *G*. The drop in potential from *a* to *c* will then be the same as from *a* to *d*, and *ac* bears the same proportion to *acb* as *ad* bears to *adb*. From this it follows that

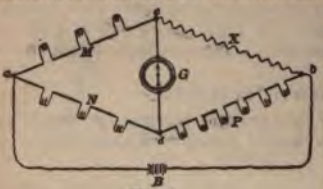


FIG. 1.

$ac : ad :: cb : db$, or the unknown resistance $X = \frac{M\,P}{N}$.

EXAMPLE.—For a certain test the ratio of the arms, $\frac{M}{N} = \frac{10}{100}$,

On adjusting the resistance P, a balance is obtained when it is equal to 7,800 ohms. Then $X = \frac{10 \times 7{,}800}{100} = 780$ ohms.

A commercial form of bridge is shown in Fig. 2. The same letters of reference are used as in the above diagram. Two keys, K and K', are added, to be used in closing the circuits.

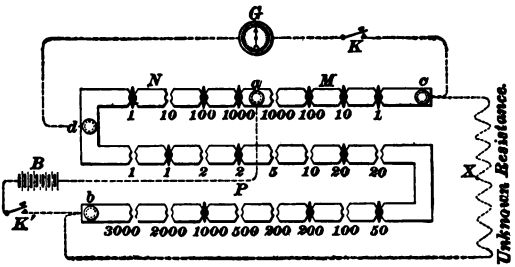


FIG. 2.

Resistances are put in by withdrawing the plugs. In the arm N there is a resistance of 10 ohms; in M, 1,000 ohms; in P, 5,888 ohms. If the galvanometer G indicates a balance, the value of the unknown resistance $X = \frac{1{,}000 \times 5{,}838}{10} = 583{,}800$ ohms.

Cable Testing.

TEST FOR CAPACITY.

A condenser of known capacity k, is charged by a battery and discharged through a galvanometer, producing a deflection d_1. The cable, having an unknown capacity k_2, is charged and discharged in similar manner, giving a deflection d_2. Then $k_2 = k_1 \frac{d_2}{d_1}$. The connections for the test are shown in Fig. 1. A plug commutator p may be used to make connection with the

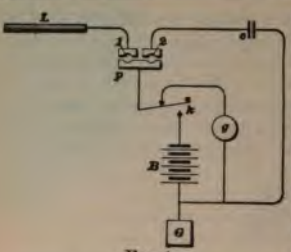


FIG. 1.

insulated line wire L or with one side of the condenser c, by putting a plug in 1 or 2. On depressing the key k, contact is made with one pole of the battery B, having about 100 cells; on releasing the key the discharge from the line or the condenser passes through the galvanometer to the ground at G.

EXAMPLE.—The deflection through a condenser of 1.5 microfarads (mfds.) was 82 divisions and through a cable, 154 divisions. Find the capacity of cable.

From the formula given,

$$k_2 = 1.5 \times \frac{154}{82} = 2.8 \text{ mfds.}$$

VOLTMETER METHOD OF TESTING INSULATION.

An ordinary Weston voltmeter with a range of 150 volts has a resistance of about 19,000 ohms. If, then, this instrument be connected across a 110-volt circuit, it will indicate the resistance of the circuit, that is, of itself, since the resistance of the armature and leads is very low. If v be the voltage across the mains, r the resistance of the voltmeter, and x the voltmeter reading, then the resistance to be determined, $R = \frac{v\,r}{x}$. When the voltmeter is put across the mains, $v = 110$, $r = 19{,}000$, and $x = 110$. The only resistance in the circuit is the voltmeter itself, for $R = \frac{110 \times 19{,}000}{110} = 19{,}000$ ohms. If we now put in series with the voltmeter a high resistance, thereby reducing the reading to 2 divisions, the total resistance $R = \frac{110 \times 19.000}{2} = 1{,}045{,}000$ ohms.

From this we must subtract the voltmeter resistance in order to

find the added resistance, which is 1,045,000 — 19,000 = 1,026,000 ohms. A deflection of one division gives 2,071,000 ohms. To obtain higher readings, a special high resistance voltmeter should be used. The connections are made as shown in Fig. 2, where V is the voltmeter, F the feeder, and D the source of current. If I = the insulation resistance of a feeder, the corrected formula becomes $I = \frac{v\,r}{x} - r$. When a voltmeter is used having a resistance of one megohm (one million ohms), then a deflection of one division, when connected up as shown, would give an insulation resistance $I = \frac{110 \times 1{,}000{,}000}{1} - 1{,}000{,}000 = 109$ megohms.

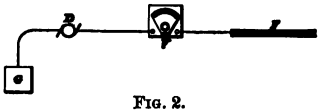


FIG. 2.

LOSS OF CHARGE METHOD OF CABLE TESTING.

The core of the cable must first be put to earth a sufficient length of time to be thoroughly cleared from any charge due to previous electrification; then the far end is freed, and connections are made as shown in Fig. 3. On depressing the key k, the cable is put to earth through the condenser c, which should be of very small capacity, say one-fiftieth of a microfarad. Both the cable L and the condenser c are then charged from the battery B by depressing the key k', and on releasing k, the condenser is discharged through the ballistic galvanometer g, a moment being chosen when the galvanometer is at zero, showing that the charge is steady. The deflection produced (d_1) represents the full charge held by the cable. The key k is *then again depressed*, and cable and condenser are charged for,

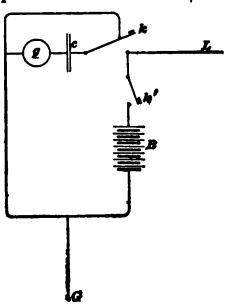


FIG. 3.

say, half a minute, after which the battery is disconnected at k', and leakage of the charge is allowed to take place for perhaps five minutes. Selecting a moment when the charge is steady, indicating an even distribution, the key k is raised, and the condenser discharged through the galvanometer. The deflection (d_2) obtained will be less than the first one, owing to the leakage of charge during the five minutes, and will therefore be a measure of the conducting power of the cable covering, or its insulation resistance. The ratio of these two deflections, d_1 and d_2, will ordinarily be sufficient to indicate the condition of the cable without further calculation; the exact insulation resistance may be found by the following formula: $I = \frac{26.06\ t}{K \log \frac{d_1}{d_2}}$,

where I is the insulation resistance of the cable in megohms, t the time in minutes during which the charge is allowed to leak, K the capacity of the cable in microfarads, d_1 the initial discharge deflection, and d_2 the final deflection after t minutes.

EXAMPLE.—In a loss of charge insulation test, the initial deflection was 238 divisions, and the deflection after 5 minutes' leakage was 137 divisions. The capacity of the cable being 1.8 microfarads, what was the insulation resistance?

$$I = \frac{26.06 \times 5}{1.8 \times \log \frac{238}{137}} = 308.1 \text{ megohms.}$$

The battery used in this test may be about 100 chloride of silver cells, or the same number of Leclanches. In the latter case it will be better to make the electrolyte of only about one-fifth the usual strength, to prevent creeping of the salts, as only very small currents are required for these tests. The battery must be very thoroughly insulated.

LOCATION OF FAULTS.

A fault in a cable usually develops slowly, and there is considerable resistance at that point; therefore, in determining the

location of the fault its resistance must be taken into account. Let *A B*, Fig. 4, be the cable, and a fault *F* connect to ground *G* through a resistance *R*. When the end *B* of the cable is insulated, the resistance is measured at the station *A*, and is equal to that portion of the cable between the station and the fault *plus* the resistance of the fault, that is, $x + R$. *B* is then grounded at *G'*, and the resistance is $x + \frac{y R}{y + R}$.

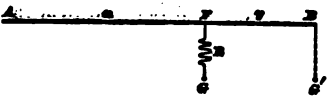


FIG. 4.

Let $x + R = r$.

Let $x + \frac{y r}{y + R} = r'$.

Let $x + y = r''$.

Then,

$$x = r' - \sqrt{(r - r')(r'' - r')}$$

$$y = r'' - r' + \sqrt{(r - r')(r'' - r')}.$$

If L = length of cable in feet, the distance from *A* to the fault $= \frac{L x}{x + y}$.

EXAMPLE.—The resistance of a cable in good condition is 3 ohms. A fault develops, and, on testing, the resistance through it is 160 ohms, the far end of the cable being insulated. When the far end is grounded, the resistance is 2.95 ohms. What is the distance to the fault, the length of cable being 5,180 feet?

$$r = 160,\ r' = 2.95,\ r'' = 3.$$

Then $x = 2.95 - \sqrt{157.05 \times .05} = 0.15$ ohm.

$$y = 3 - 2.95 + \sqrt{157.05 \times .05} = 2.85 \text{ ohms.}$$

The distance to the fault $= \frac{5.180 \times .15}{3} = 259$ feet.

Bell Wiring.

The simple bell circuit is shown in Fig. 1, where p is the push-button, b the bell, and c, c the cells of the battery connected up in series. When two or more bells are to be rung from one push-button, they may be joined up in parallel across the battery wires as in Fig. 2 at a and b, or they may be arranged in series as in Fig. 3. The battery B is indicated in each diagram by short parallel lines, this being the conventional method. In the parallel arrangement of the bells, they are independent of each other, and the failure of one to ring would not affect the others; but in the series grouping all but one bell must be changed to a single stroke action, so that each impulse of current will produce only one movement of the hammer. The current is then interrupted by the vibrator in the remaining bell, the result being that each bell will ring with full power. The only change necessary to produce this effect is to cut out the circuit-breaker on all but one bell by connecting the ends of the magnet wires directly to the bell terminals.

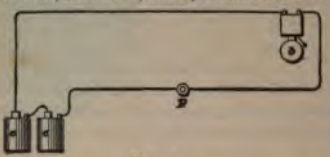


FIG. 1.

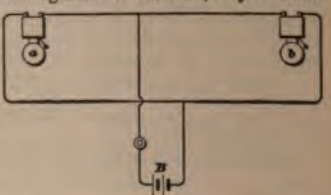


FIG. 2.

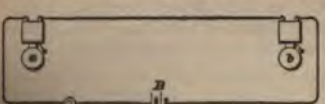


FIG. 3.

When it is desired to ring a bell from one of two places some distance apart, the wires may be run as shown in Fig. 4. The points p, p' are located at the required points, and the battery

and bell are put in series with each other across the wires joining the pushes.

A single wire may be used to ring signal bells at each end of a line, the connections being given in Fig. 5. Two batteries are required, B and B' and a key and bell at each station. The keys k, k' are of the double contact type, making connections normally between the bell b or b' and the line wire L. When one

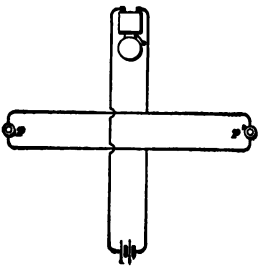


FIG. 4.

key, as k, is depressed, a current from B flows along the wire through the upper contact of k' to the bell b' and back by way of the ground-plates G', G.

When a bell is intended for use with burglar alarm apparatus,

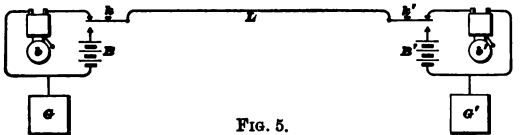


FIG. 5.

a constant-ringing attachment may be introduced, which closes the bell circuit through an extra wire as soon as the trip at door or window is disturbed. In the diagram, Fig. 6, the main circuit, when the push p is depressed, is through the automatic drop d by way of the terminals a, b to the bell and battery. This current releases a pivoted arm which, on falling, completes the circuit between b and c, establishing a new path for the current by way of e, independent of the push p.

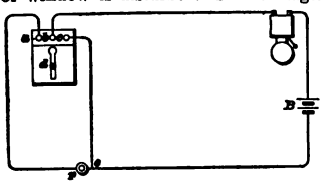


FIG. 6.

Annunciator Systems.

The wiring diagram for a simple annunciator system is shown in Fig. 1. The pushes *1*, *2*, *3*, etc., are located in the various rooms, one side being connected to the battery wire *b*, and the other to the leading wire *l* in communication with the annunciator drop corresponding to that room. A battery of two or three Leclanche cells is placed at *B* in any convenient location. The size of wire used throughout may be No. 18 annunciator wire.

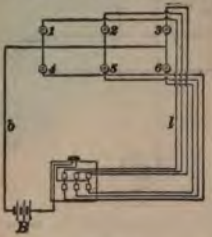


FIG. 1.

A return call system is illustrated in Fig. 2, in which there is one battery wire *b*, one return wire *r*, and one leading wire l, l_2, etc. for each room. The upper portion of the annunciator board is provided with the usual drops, and below these are the return call pushes. These are double contact buttons, held normally against the upper contact by a spring. When in this position, the closing of the circuit by the push-button in any room, such as No. 4, rings the office bell and releases No. 4 drop, the path of the current in this case being from push 4 to *a-c-d-e-f-g-B-h-b* back to the push-button. On the return signal being made by pressing the button at the lower part of the annunciator board, the office bell circuit is broken at *d*, and a new circuit formed through *k* as

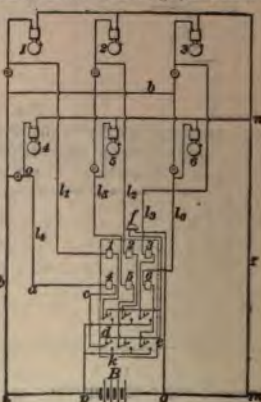


FIG. 2.

follows: From the battery *B* to *g-m-r-n-o-a-c-k-p* to battery, the room bell being in this circuit. A general fire alarm may be added to this system, consisting of an automatic clockwork apparatus for closing all the room bell circuits at once, or as many at a time as the battery can ring. When this system is installed, the battery wire should be No. 14 or 16. Four or five Leclanche cells are usually required in this case.

It will be seen that the connections are so arranged that the room bell will ring when the push in that room is pressed. If this be not desired, a double contact push may be substituted, so that the room bell circuit is broken at the same time that the circuit is made through the annunciator. This double push should be so connected that the circuit is normally complete through the bell, the leading wire being connected to the tongue and the battery wire to the second contact point, normally out of circuit.

EXTRACT FROM THE REGULATIONS OF THE UNDERWRITERS' ASSOCIATION.

Incandescent Wires.

Conducting wires, carried over or attached to buildings, must be (a) at least seven feet above the highest point of flat roofs, and (b) one foot above the ridge of pitch roofs; (c) when in proximity to other conductors likely to divert any portion of the current, they must be protected by guard irons or wires, or a proper additional insulation, as the case may require.

For entering buildings (a) wires with an extra heavy waterproof insulation must be used; (b) they must be protected by drip loops; (c) also protected from abrasion by awning frames: (d) be at least six inches apart; (e) the holes through which they pass in the outer walls of such buildings must be bushed with a non-inflammable, waterproof, insulating tube, and (f) should slant upward toward the inside.

(a) Wires must never be left exposed to mechanical injury, or to disturbance of any kind. (b) Wires must not be fastened by metallic staples. (c) When wires pass through walls, floors, partitions, timbers, etc., glass tubing, or so-called "floor in-

Dynamo Machines.

Dynamo machines must be located in dry places, not exposed to flyings or easily combustible material, and insulated upon wooden foundations. They must be provided with devices capable of controlling any changes in the quantity of the current; and if the governors are not automatic, a competent person must be in attendance near the machine whenever it is in operation.

Each machine must be used with complete wire circuits; and connections of wires with pipes, or the use of circuits in any other method are absolutely prohibited.

The whole system must be kept insulated, and tested *every day* with a magnet for ground connections in ample time before lighting, to remedy faults of insulation, if they are discovered; and proper testing apparatus must in each case be provided. This applies to both central station and isolated plants.

Testing circuits for grounds with a battery and bell is not considered a reliable test.

Preference is given to switches constructed with a lapping connection, so that no electric arc can be formed at the switch when it is changed; otherwise the stands of switches, where powerful currents are used, must be made of some incombustible substances which will withstand the heat of the arc when the switch is changed.

Motors.

Wires for motors should be run exactly as for lamps on similar circuits.

On low-tension circuits, where motors are run in multiple, safety catches must be used on each side of the circuit.

On high-tension circuits the same restrictions apply as for arc lamps, and suitable cut-outs must be provided.

Motors must be treated as dynamos as regards insulation, flyings, dampness, etc.

NOTE.—If the regulations of the Underwriters' Association are not followed in wiring buildings, the wiring is liable to be condemned by the Insurance Inspectors and the policy cancelled.

ARCHITECTURE.

Thickness of Walls Required in New York City.

DWELLING HOUSES, APARTMENT HOUSES, HOTELS AND SCHOOLS.

Height of Walls. *Measured from the curb at center of building.*	*Foundations*		*Outside and Bearing Walls.*
	Stone.	*Brick*	
	in.	*in.*	
Not exceeding 35 ft.....	20	16	12 in. in basement and 8 in. above.
35 ft. and not over 50 ft.	20	16	12 in. above foundation.
50 " " " 60 "	24	20	16 in. in first story, if level with ground, and 12 in. above.
60 " " " 75 "	24	20	16 in. for first 25 ft. and 12 in. above.
75 " " " 85 "	28	24	20 in. for first 20 ft., 16 in. for the next 40 ft., and 12 in. above.
85 " " " 100 "	32	28	24 in. for the first 35 ft., 20 in. for the next 35 ft., and 16 in. above.
100 " " " 115 "	36	32	28 in. for the first 25 ft., 24 in. for the next 25 ft., 20 in. for the next 40 ft., and 16 in. above.

Non-bearing partition walls under 50 ft. in height may be 8 in. in thickness. The height in all cases to be taken to the nearest tier of floor beams. Non-bearing walls may be 4 in. less in thickness than is given in the above table, but are not to be less than 12 in.

Bearing walls for dwelling and apartment houses, hotels and schools, exceeding 115 ft. in height, should be increased 4 in. at the bottom for every additional 25 ft. in height or fraction thereof, the upper 115 ft. remaining the same as given for a wall of that height in the above table.

Thickness of foundation walls given in table is for a depth not exceeding 120 in. below the curb; for every additional 10 ft. in depth or part thereof they shall be increased 4 in. in thickness.

Height of Walls. Measured from the curb at center of building.	*Foundations* Stone.	*Foundations* Brick	*Outside and Bearing Walls.*
	in.	*in.*	
Not exceeding 40 ft.....	20	16	12 in. above foundation.
40 ft. and not over 60 ft...	24	20	16 in. for first 40 ft. and 12 in. above.
60 " " " 75 " ..	28	24	20 in. for first 25 ft. and 16 in. above.
75 " " " 85 " ..	32	28	24 in. for first 25 ft., 20 in. for the next 35 ft., and 16 in. above.
85 " " " 100 " ..	36	32	28 in for first 25 ft., 24 in. for the next 25 ft., 20 in. for the next 25 ft., and 16 in. above.

If there is to be a clear span exceeding 25 ft. between the bearing walls, they shall be 4 in. more in thickness than is given in the above table for every 12½ ft., or fraction thereof over 25 ft.

Bearing walls for warehouses exceeding 100 ft. in height should be increased 4 in. at the bottom for every additional 25 ft., or fraction thereof, the upper 100 ft. remaining the same as given for such wall in the above table.

Floor Loads Exclusive of Weight of Construction.

Floors of dwellings, hotels, etc	70	lbs. per sq. ft.
" churches, theatres and ball rooms	80 to 120	" "
" schools	80	" "
" hay lofts	80	" "
Floors for storage of grain	100	" "
Floors of warehouses and general merchandise	150 to 400	" "
" factories	150 to 250	" "
" office buildings	100	" "

Bearing Power of Soils in Tons per Square Foot.

Kind of Material.	*Min.*	*Max.*
Rock, hardest in native bed	200	
" equal to best ashlar masonry	25	30
" " " brick	15	20
" " poor "	5	10
Clay, dry in thick beds	4	6
" moderately dry in thick beds	2	4
" soft	1	2
Gravel and coarse sand well cemented	8	10
Sand, compact and well cemented	4	6
" clean, dry	2	4
Quick sand, alluvial soils, etc	0.5	1

Working Strength of Brick Work.

	Tons per sq. ft.
Eastern hard burnt bricks in lime mortar	8 to 10
" " " " 1 part of Rosendale cement and 2 parts of sand	12
" " " " 1 part of Rosendale cement and 3 parts of lime mortar	14
" " " " 1 part of Portland cement and 2 parts of sand	15
Western hard burnt bricks in Louisville cement	10
" " " " 1 part of Portland cement and 2 parts of sand	12½

Piers should be well built, bonded and have iron or stone caps.

Drilling and Blasting in Granite.

In drilling and wedging blocks of granite so that they can be reduced to dimensions for building purposes, an average day's work of a man working ten hours is to drill 50 holes ⅝-in. in diameter and 3 in. deep, put in the plugs and feathers and split the rock. If 1-in. diameter holes 3½ in. deep are required, as *Lewis* holes, one man will drill 20 holes in a day of ten hours.

In drilling rock in the quarry for blasting purposes, three men will sink a hole 10 ft. deep and from 2 in. to 2½ in. in diameter; one man holding the drill and two men striking.

In marble, limestone and sandstone from 20 to 30 per cent. more depth can be drilled.

A steam drill in one day will drive from 70 to 90 feet according to circumstances, the hole being from 2 in. to 2½ in. in diameter. If it is required to sink a hole 20 ft. deep it will be necessary to start it 3 in. in diameter, as the drill will lose ¼ in. in each 5 ft. drilled.

The explosive force of powder is about 18 tons per square inch, its weight is about the same as water, which is 62½ lb. per cubic foot, therefore, 1 lb. will be equal to about 28 cubic inches.

In ordinary open quarrying, a cubic yard of solid rock will require from ¼ to 1 lb. of powder.

In tunnels and shafts, from 1½ to 5 lb. per cubic yard. Where the rock is soft and shaly, it frequently requires more powder than solid rock.

It is usual to prime or enlarge the holes by a small charge of dynamite before the full charge of powder is placed in the hole.

Nitro-glycerine will explode by concussion or under pressure exceeding 2,000 pounds per square inch if the pressure is suddenly applied, or at a temperature of 600°. Tamping is unnecessary when nitro-glycerine is used in blasting. Weight for weight, nitro-glycerine has from 8 to 10 times the explosive force of powder.

Dynamite is composed of 75 parts of nitro-glycerine and 25 parts of siliceous earth; it is insoluble in water, it may be used in wet holes, and it requires no tamping. It congeals at 40° F., and is rendered ineffectual at 212° F. Its explosive force is from 3 to 4 times that of powder.

Concrete Work for Foundations and Floors.

Only the best brands of Portland cement should be used for these purposes, and the proportions may be as follows:

One part of cement, 3 parts of clean, sharp sand and 5 parts of stone, broken small enough to pass through a two-inch ring. In preparing the concrete, the sand and broken stone must be first carefully washed clean immediately before being used; then the cement, sand and broken stone are mixed together, and an additional amount of water added by means of a spray, to form a pasty matrix, the whole being well worked over and incorporated, and immediately deposited in position; it must be well tamped. In heavy work, the layers should not exceed 12 in. in thickness, and should be allowed to set for 24 hours before the succeeding layer is deposited. For making 1 cubic yard of concrete mixed in the above proportions, the following quantities of material will be required:

1 Barrel of cement,
⅙ Cubic yard or .6 ton of sand,
1 Cubic yard or 1 ton of broken stone.

Basement floors are usually finished with a ½-inch coat of imported cement in the proportion of 2 parts cement to 1 part of fine, clean, white sand, and finished with a floated surface; but for heavy work, the coat should be 1 inch in thickness.

A coat 1 inch thick will require for each superficial yard .182 barrel of cement and .091 barrel of sand. If ½ inch thick, one-half the quantity will be required.

The body of the concrete should *not* be allowed to dry before the finishing coat is applied.

MEMORANDA ON MASONRY.

SIZES OF BRICKS.

	Thickness.		*Width.*		*Length.*
Common brick	From 2 to 2¼	×	4	×	8¼ inches
Philadelphia pressed	2¼	×	4	×	8⅝ "
Peerless Brick Co.'s make	2⅜	×	4⅛	×	8⅝ "
Baltimore and Trenton	2⅜	×	4⅛	×	8¼ "
Croton	2¼	×	4	×	8½ "
Colabaugh	2⅜	×	3⅝	×	8¼ "
Maine common	2⅜	×	3⅜	×	7½ "
North River common	2¼	×	3½	×	8 "
Milwaukee	2⅜	×	4⅛	×	8¼ "
Stourbridge Fire Brick	2⅜	×	4⅜	×	9⅛ "

BRICKWORK.

Brickwork may be computed either by cubic measurement or by the method in general use, on a rate per 1,000 bricks laid in the wall. Where common bricks measuring $2 \times 4 \times 8\frac{1}{4}$ inches are used, having mortar joints $\frac{1}{4}$ inch thick, the number required per cubic yard will be 574, and per cubic foot, 21.26. The proportion of mortar to the whole mass will be one-fourth.

The average number of bricks required per square foot of wall surface for different thicknesses of wall will be in accordance with the following table:

Thickness of Wall.	*No. of Bricks Required.*
1 Brick or 4 inches	7
2 Bricks or 8¼ inches	14
1½ " " 12½ "	21
2 " " 16¾ "	28
2½ " " 21 "	35
3 " " 25¼ "	42

An allowance of five per cent. should be added to the amount to compensate for waste.

The weight of common brick averages 4.5 lbs. each or 118 lbs. per cubic foot; this will give 3,186 lbs. or 1.42 long tons per cubic yard or 2.01 tons per 1,000 bricks.

The weight of pressed brick of the same dimensions as the common brick will be about 5 lbs., or 131 lbs. per cubic foot; 3,537 lbs. or 1.58 tons per cubic yard or 2.23 tons per 1,000 bricks.

Brickwork constructed with common brick having $\frac{5}{8}$-inch joints weighs 112 lbs. per cubic foot, and with pressed brick having $\frac{1}{8}$-inch joints, 127 lbs. per cubic foot.

MORTAR, ETC.

Lime	1 Bushel =	66 lbs. = 1.25 cubic feet.
	1 Cu. ft. =	53 "
	1 Load =	20 bushels or 25 cubic feet.
Sand	1 Bushel =	122 lbs. = 1.25 cubic feet.
	1 Cu. ft. =	95 lbs.
	1 Load =	20 bushels or 25 cubic feet.

A one-horse load both in the United States and in Great Britain usually means a cubic yard, but many dealers have adopted 20 struck bushels, or 25 cubic feet as a load.

Mortar mixed in the proportion of 1 measure of *quicklime*, either in lump or ground, to 5 measures of sand, and tempered with clean water furnishes an excellent compound, and is as good as can be desired for the purpose, provided that the lime is fresh and the sand clean, sharp and free from loam and other impurities.

The lime in slaking doubles in bulk, so that the proportion of *slaked lime* to the sand in mortar would be 1 slaked lime to 2½ of sand. The bulk of the mortar will be about ⅛ greater than that of the dry sand alone, so that 20 cubic feet or 16 bushels of sand, and 4 cubic feet or 3.2 bushels of quicklime, will make about 22½ cubic feet of mortar.

With lime at 35 cents per bushel and sand at 5 cents per bushel, the *cost* of the mortar without the labor will be 8.6 cents per cubic foot.

For laying 1,000 bricks with ¼-inch joints the cost of the mortar at the above rate would be $1.00. For ordinary brick-work such as interior walls of houses, where the joints are from ⅜ inch to ½ inch, the cost of the mortar for 1,000 bricks would be about $1.60, one-third of the mass being mortar.

For Rubble Masonry, in which one-third of the mass may be considered mortar, there would be 9 cu. ft. at 8.6 cents = 77.4 cents per cubic yard. One barrel of lump lime and ¾ of a load of sand will make sufficient mortar for 1,000 bricks in ordinary work, or for 2 perches of 25 cubic feet of Rubble Masonry.

The weight of mortar when dry is about 103 lbs. per cu. ft.

BRICK DUST MORTAR.

Experiments recently made with mixtures of brick dust and quicklime in blocks of ½ inch in thickness, showed that after immersion in water for some time, they withstood, without crushing, a pressure of 1,500 lb. per squre inch.

The addition of a small proportion of brick dust to ordinary mortar is deemed a preventive of disintegration so common to mortar exposed to the action of the elements.

The mixture of brick dust with lime and sand is said to be generally employed in the Spanish Dominions and considered *superior to hydraulic* cement for drains, cisterns, etc., also for *roofs and in setting* tiles.

The proportions used are one of brick dust, one of lime and two of sand, mixed together, dry and then tempered with water in the usual way.

DURABLE WHITE WASH.

A whitewash adopted by the United States Government and used for coating light-houses and keepers' dwellings, is composed as follows:

"To ten parts of freshly slaked lime add one part of best hydraulic cement. Mix well with salt water."

This white wash when properly mixed and applied, produces a clear white that does not easily *rub* or wash off.

WATERPROOFING OF BRICK WALLS.

A process employed for waterproofing brick walls consists of two solutions applied alternately, one being of soap and water and one of alum and water.

Dissolve ¾ lb. soap in 1 gallon of water, and ½ lb. of alum in 4 gallons of water.

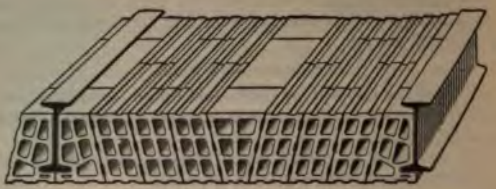
The first or soap solution should be at boiling point, and the walls clean and dry. Apply with flat brush, taking care to form no bubbles on surface of brick; allow this coat to remain 24 hours. Apply alum solution, which should be about lukewarm, and allow this also to remain 24 hours. Continue the operation alternately with the two solutions until the walls are found to be impervious to moisture.

The alum and soap form an insoluble compound which fills the pores of brick and prevents absorption of moisture. Two coats of each solution is considered sufficient for brick walls under ordinary conditions.

BRICKWORK.

Efflorescence upon the surface of brick walls laid in mortar of which natural hydraulic lime or cement is the basis can be prevented by mixing with the mortar .025 of its weight of animal fat.

Crystallization of the salts within the pores of the bricks, into which they have been absorbed from the mortar, causes disintegration of the brick and a consequent weakening of the wall.



HOLLOW BRICK FOR FLAT ARCHES.

Width of Span between Iron Beams.	*Depth of Arch.*	*Weight per sq. foot.*	*Safe load per sq. foot.*
3 ft. 6 in. to 4 ft. 0 in.	6 in.	29 lbs.	1,500 lbs.
4 ft. 0 in. to 5 ft. 0 in.	7 "	33 "	"
4 ft. 6 in. to 6 ft. 0 in.	8 "	37 "	"
5 ft. 0 in. to 7 ft. 0 in.	9 "	40 "	"
5 ft. 0 in. to 7 ft. 6 in.	10 "	43 "	"
6 ft. 0 in. to 8 ft. 0 in.	12 "	48 "	"

Partitions.	*Thickness.*	*Weight per square foot.*
Hollow Brick (Clay)	3 in.	14 lbs.
" " "	4 "	18½ "
" " "	5 "	23 "
" " "	6 "	25 "
" " "	7 "	31 "
" " "	8 "	34 "
Porous Terra-Cotta	3 "	12 "
" " "	4 "	17 "
" " "	5 "	23 "
" " "	6 "	27 "
" " "	7 "	31 "
" " "	8 "	36 "

HOLLOW BRICK FOR FLAT ARCHES.—*Continued.*

Furring, Roofing and Ceiling.	*Thickness.*	*Weight per square foot.*
Hollow Clay Furring.....	2 in.	12 lbs.
Porous Terra-Cotta Furring.....	2 "	8 "
" " " Roofing.....	2 "	12 "
" " " "	3 "	16 "
" " " Ceiling.....	2 "	11 "

A 5-foot span (18 in. wide) of 8-in. Floor Block is found, by experiment, to stand 9,300 lbs. without breaking.

COMPARATIVE COST OF WOOD AND FIREPROOF CONSTRUCTION.

WOOD.

Description.			*Weight per sq. foot in lbs.*				*Cost per sq. foot in cents.*			
Floor Space.	*Size of Joists.*	*Spacing Ctr. to Ctr.*	*Joists and Bridging.*	*Flooring.*	*Plaster and Furring.*	*Total.*	*Joist and Bridging in place.*	*Plaster and Furring in place.*	*Flooring in place.*	*Total.*
10 × 10	2 × 8	12	4	15	9	28.	3.5	3	11.5	18.
12 × 12	2 × 9	12	4	15	9	28.	4.	3	11.5	18.5
14 × 14	2 × 10	12	4.5	15	9	28.5	4.	3	11.5	18.5
16 × 16	2 × 12	12	5	15	9	29.	4.5	3	11.5	19.
18 × 18	3 × 10	12	6.33	15	9	30.33	7.	3	11.5	21.5
20 × 20	3 × 12	12	7.7	15	9	31.7	6.5	3	11.5	21.
22 × 22	4 × 10	12	8	15	9	32.	7.33	3	11.5	21.83
24 × 24	4 × 12	12	9.75	15	9	33.75	9.	3	11.5	23.5
26 × 26	4 × 16	15	10.66	15	9	34.66	8.5	3	11.5	23.

Based on labor at $3.50 per day.
Fiber strain 1,000 lbs. per sq. inch.
Load 100 to 125 lbs. per sq. foot.

FIREPROOF LOAD OF 125 LBS. PER SQ. FOOT.

Description.			Weight per sq. foot in lbs.				Cost per sq. foot in cents.			
Floor Space.	Steel Beams.	Spacing.	Steel Construction.	Arches, Flooring, etc.	Plaster.	Total.	Steel Construction in place.	Arches, etc. in place.	Plaster in place.	Total.
10 × 10	6″—1 31b.	5′	3.	42	6	51.	6.75	24.	3	33.75
12 × 12	7″—1 51b.	6′	3.	44	6	53.	6.75	24.5	3	34.25
14 × 14	7″—1 51b.	4′ 8″	4.	44	6	54.	9.	24.5	3	36.50
16 × 16	9″—2 1 lb.	5′ 4″	4.5	52	6	62.5	10.25	28.	3	41.25

FIREPROOF LOAD OF 150 LBS. PER SQ. FOOT.

18 × 18	10″—30lb.	6′	5.	56	6	68.	13.5	28.	3	44.5
20 × 20	12″—32lb.	6′ 8″	6.	57	6	79.	13.5	30.	3	46.5
22 × 22	12″—40lb.	5′ 6″	8.	67	6	81.	18.	30.	3	51.
24 × 24	12″—40lb.	4′ 9″	9.5	67	6	82.5	21.25	30.	3	54.75
26 × 26	12″—50lb.	5′ 3″	10.5	67	6	83.5	23.5	30.	3	56.5

Fibre stress = 16,000 lbs. sq. inch.

This table does not include cost of columns or partitions, but simply the floor construction itself. The cost, including columns, would vary with the height of the story and the number of stories in the building. Interior non-supporting partitions composed of T iron and 4″ fireproof blocks and plastered both sides would cost 16 cents per square foot of area if built when the building is erected, or from 20 cents to 24 cents per square foot if constructed after the building is complete.

Transverse Strength of Flagging.

b = width of the stone in inches.

d = its thickness in inches.

l = distance between bearings in inches.

The following gives the *breaking loads* in tons of 2,000 lbs. Weight being placed at the center of the span.

$$\frac{b\,d^2}{l} \times \left\{ \begin{array}{lr} \text{For} & \\ \text{Blue Stone Flagging} & .744 \\ \text{Quincy Granite} & .624 \\ \text{Little Falls Freestone} & .576 \\ \text{Belleville, N. J., Freestone} & .480 \\ \text{Connecticut Freestone} & .312 \\ \text{Dorchester Freestone} & .264 \\ \text{Aubigny Freestone} & .216 \\ \text{Caen Freestone} & .144 \\ \text{Glass} & 1.000 \\ \text{Slate} & 1.2 - 2.7 \end{array} \right.$$

Thus a block of Quincy Granite 80 inches wide and 6 inches thick, resting on supports 36 inches in the clear, would be broken by a load resting midway between supports of $\frac{80 \times 36}{36} \times .624$ = 49.92 tons.

For safe load divide the result by 10.

Average weight of flagging per cubic foot = 168 lbs. Weight per square foot as follows :

Thickness in inches	1	2	3	4	5	6	7	8
Weight in lbs	14	28	42	56	70	84	98	112

Cost of 100 Sq. Yds. 3 Coats Plaster, New York City.

Common lime	4 bbls.	$4.00
Finishing lime	¾ "	.88
Plaster Paris	⅛ "	.75
Hair	4 bushels,	.86
Sand (common)	7 loads,	2.10
" (white)	2½ bushels,	.25
Nails	12½ lbs.,	.75
Lath	2,000,	4.00
Plasterer, 3 days @	$3.00,	9.00
Laborer, 2 " @	1.00,	2.00
		$24.59

ROOFS.

Angles of Roofs.

Proportion of Rise to Span.	*Angle.*	*Slope.*	*Proportion of Rise to Span.*	*Angle.*	*Slope.*
$\frac{1}{5}$	21° 50′	2½ to 1	$\frac{2}{3}$	53° 10′	¾ to 1
$\frac{1}{4}$	26° 30′	2 " 1	$\frac{3}{4}$	56° 20′	$\frac{2}{3}$ " 1
$\frac{1}{3}$	33° 40′	1½ " 1	1	63° 30′	½ " 1
$\frac{1}{2}$	45° 00′	1 " 1			

WEIGHT OF ROOF COVERINGS IN POUNDS PER SQ. FOOT.

	Lbs.
Slate, $\frac{3}{16}$" thick, on one-inch boards	10.0
Slate, $\frac{1}{8}$" " " " " "	7.5
Corrugated iron, No. 20, on one-inch boards	6.0
Felt on boards, 3 ply, " " " "	3.5
Tin on one-inch boards	4.0
Slate on T purlins	12.0
Corrugated iron and laths	6.0
Slate or iron laths	10.0

When slate is used on purlins of T iron, the purlins should be 2" × 1" × ¼", 10 pounds per yard, and spaced from 10" to 12" apart, the spacing between rafters (jacks and principals, or between jacks and jacks) should be about 5 feet.

COLUMNS AND GIRDERS.

Ultimate Strength of Rectangular Wooden Pillars Well Seasoned.

Let l = length of pillar in feet.
d = least width of side in inches.

Green timber has about half the strength of dry.

In the following table $\frac{l}{d}$ is the ratio of length of post to least

side of cross-section and f the ultimate resistance to compression, in pounds per square inch. To obtain the safe resistance, divide the ultimate resistance by 6.

White Pine.			*Georgia Pine.*		
$\frac{l}{d}$	f	*Ratio of Decrease.*	$\frac{l}{d}$	f	*Ratio of Decrease.*
0 to 10	2,500	Base.	0 to 15	4,000	Base.
10 " 35	2,000	less 20%	15 " 30	3,500	less 12½
35 " 45	1,500	" 40%	30 " 40	3,000	" 25%
45 " 60	1,000	" 60%	40 " 45	2,500	" 37½
			45 " 50	2,000	" 50%
			50 " 60	1,500	" 62½

Approximate values of from 75 to 80 per cent. of those given for Georgia Pine may be assumed for White Oak posts.

Girders.

The condition of stress sustained by a girder is that of a beam supported at both ends and carrying a load evenly distributed, or concentrated at one or more points throughout its length.

To compute the dimensions of an evenly loaded wooden girder use the following formulas:

$$\text{Breadth in inches} = \frac{3\,L \times W}{4\,F\,D^2} \qquad \text{Depth in inches} = \sqrt{\frac{3\,L \times W}{4\,F\,B}}$$

$$\text{Safe load in pounds} = \frac{4\,F\,B\,D^2}{3\,L}$$

B = breadth in inches.
D = depth " "
F = safe fibre stress.
L = distance between supports in inches.
W = safe load.

EXAMPLES:

What breadth of pine girder 12" deep will be required to carry a distributed load of 4,000 pounds when the span is 18 ft. and the fibre stress is not to be more than 800 lb. per sq. in.?

$$\text{Breadth} = \frac{3\,L \times W}{4\,F\,D^2} = \frac{3 \times 216 \times 4000}{4 \times 800 \times 12^2} = \frac{2592000}{460800} = 5.62.$$

Therefore, a girder 6" × 12" would be required. When the girder has its load concentrated at the center as is the case when it carries at its center a column or post, it will carry but half the load it is capable of sustaining when the load is evenly distributed. The formulas for girders when the load is concentrated at the center are

$$\text{Breadth in inches} = \frac{6\,L\,W}{4\,F\,D^2}$$

$$\text{Depth in inches} = \sqrt{\frac{6\,L\,W}{4\,F\,B}}$$

$$\text{Safe load in pounds} = \frac{4\,F\,B\,D^2}{6\,L}.$$

When the girder has a concentrated load at some point not at the center, it will carry more than if loaded at the center. The strength of the beam is then expressed in the following ratio; $m\,n : \left(\frac{L}{2}\right)^2 :: \frac{W}{2} : W$ when m = the distance of the concentrated load from one end of the beam. n = the distance of the concentrated load from the other end. W = the safe load at the point indicated, and the other letters have the same values as before. The formulas for girders when the load is concentrated at some point away from the center are

$$\text{Safe load in pounds} = \frac{F\,B\,D^2\,L}{6\,m\,n}$$

$$\text{Breadth in inches} = \frac{6\,W\,m\,n}{F\,D^2\,L}$$

$$\text{Depth in inches} = \sqrt{\frac{6\,W\,m\,n}{F\,B\,L}}.$$

In accordance with the Building Laws of New York, Chicago and Boston.

FORMULA: $W = \frac{4 F B D^2}{3 L}$

W = Safe Load. F = Maximum Fibre Stress.
B = Breadth, in inches. D = Depth, in inches.
L = Distance between supports, in inches.

New York.	*Chicago.*	*Boston.*
F = 800 lbs. per sq. in. for Hemlock. 900 lbs. per sq. in. for Spruce or White Pine. 1,100 lbs. per sq. in. for Georgia Pine or Oak.	F = 900 lbs. per sq. in. for White Pine. 1,080 lbs. per sq. in. for Oak. 1,440 lbs. per sq. in. for Georgia Pine.	F = 750 lbs. per sq. in. for Spruce. 1,000 lbs. per sq. in. for Oak. 1,250 lbs. per sq. in. for Georgia Pine.
EXAMPLE: Given Georgia Pine Beam 4″ × 12″ × 15′ .0″ long. $W = \frac{4 \times 1100 \times 4 \times 144}{3 \times 15 \times 12} =$ 4693 lbs.	EXAMPLE: Given Georgia Pine Beam 4″ × 12″ × 15′ .0″ long. $W = \frac{4 \times 1440 \times 4 \times 144}{3 \times 15 \times 12}$ = 6144 lbs.	EXAMPLE: Given Georgia Pine Beam 4″ × 12″ × 15′ .0″ long. $W = \frac{4 \times 1250 \times 4 \times 144}{3 \times 15 \times 12}$ = 5333 lbs.

Safe Loads Uniformly Distributed for Rectangular Spruce or White Pine Beams, One Inch Thick.

The following table has been calculated for a maximum fibre stress of 750 lbs. per sq. in. : taking a safety factor of 4.

To obtain the safe load for any thickness : Multiply values for 1 inch by thickness of beam.

To obtain the required thickness for any load : Divide by safe load for 1 inch.

For oak, increase values in table by $\frac{1}{6}$.

For Georgia pine, increase values in table by $\frac{3}{8}$.

Span in Feet.	*Depth of Beam.*										
	6"	7"	8"	9"	10"	11"	12"	13"	14"	15"	16"
5	600	820	1070	1350	1670	2030	2400	2820	3270	3750	4270
6	500	680	890	1120	1390	1680	2000	2350	2730	3120	3560
7	430	580	760	960	1190	1440	1710	2010	2330	2680	3050
8	380	510	670	840	1040	1260	1500	1760	2040	2340	2670
9	330	460	590	750	930	1120	1330	1560	1810	2080	2370
10	300	410	530	670	830	1010	1200	1410	1630	1880	2130
11	270	370	490	610	760	920	1090	1280	1490	1710	1940
12	250	340	440	560	690	840	1000	1180	1360	1560	1780
13	230	310	410	520	640	780	930	1080	1260	1440	1640
14	210	290	380	480	590	720	860	1010	1170	1340	1530
15	200	270	360	450	560	670	800	940	1090	1250	1420
16	190	260	330	420	520	630	750	880	1020	1180	1330
17	180	240	310	400	490	590	710	830	960	1100	1260
18	170	230	290	370	460	560	670	780	910	1040	1190
19	160	210	280	360	440	530	630	740	860	990	1130
20	150	200	270	340	420	510	600	710	820	940	1070
21	140	190	260	320	390	480	570	670	780	890	1020
22	140	190	240	310	380	460	540	640	740	850	970
23	130	180	230	290	360	440	520	610	710	810	920
24	130	170	220	280	350	420	500	590	680	780	890
25	120	160	210	270	330	410	480	560	660	750	860
26	110	160	210	260	320	390	460	540	630	720	820
27	110	150	200	250	310	370	440	520	610	690	790
28	110	140	190	240	300	360	430	500	580	670	760
29	110	140	180	230	290	350	410	490	560	640	740

Properties of Timber.

Description.	Weight per cubic foot in lbs.	Weight per ft. B. M. in lbs., average.	Tensile strength per sq. in., in lbs.	Crushing strength per sq. in., in lbs.	Relative strength for cross breaking. White Pine = 100.	Shearing strength with grain, lbs. per sq. in.	Pressure in lbs. per sq. in. to indent $\frac{1}{10}$".
Ash	43 to 55.8	4.1	11,000 to 17,207	4,400 to 9,363	130 to 180	458 to 700	1,800 to 1,850
Beech	43 to 53.4	3.9	11,500 to 18,000	5,800 to 9,363	100 to 144	....	....
Cedar	50 to 56.8	4.5	10,300 to 11,400	5,600 to 6,000	55 to 63	....	....
Cherry	....	....	....	....	130	....	....
Chestnut	33	2.75	10,500	5,350 to 5,600	96 to 123	....	....
Elm	34 to 36.7	2.9	13,400 to 13,489	6,831 to 10,331	96	....	....
Hemlock	....	....	8,700	5,700	88 to 95	....	....
Hickory	....	....	12,800 to 18,000	8,925	150 to 210	....	....
Locust	44	3.7	20,500 to 24,800	9,113 to 11,700	132 to 227	....	....
Maple	49	4.1	10,500 to 10,584	8,150	122 to 220	367 to 647	1,700 to 1,900
Oak, White	45 to 54.5	4.1	10,253 to 19,500	4,684 to 9,509	130 to 177	752 to 966	2,300 to 3,550
Oak, Live	70	5.8	....	6,850	155 to 189	....	....
Pine, White	30	2.5	10,000 to 12,000	5,000 to 6,650	100	225 to 423	875 to 1,160
Pine, Yellow	28.8 to 33	2.6	12,600 to 19,200	5,400 to 9,500	98 to 170	286 to 415	1,900
Spruce	....	....	10,000 to 19,500	5,050 to 7,850	86 to 110	253 to 374	875 to 1,025
Walnut, Black	42	3.5	9,286 to 16,000	7,500	....	....	2,300 to 2,600

Relative Hardness of American Woods.

If we take Shell-bark Hickory and assume a value
it, other woods will compare with it in hardness as fol

Shell-bark hickory	100	Yellow oak	
Pignut hickory	96	White elm	
White oak	84	Hard maple	
White ash	77	Red cedar	
Dogwood	75	Wild cherry	
Scrub oak	73	Yellow pine	
White hazel	72	Chestnut	
Red oak	69	Yellow poplar	
White beech	65	White birch	
Black walnut	65	Butternut	
Black birch	62	White pine	

WINDOWS.

Lengths of Round and Square Cast Iron Sash W

Diameter.	*Length of 1 lb. in inches.*	*Length of 1 lb. in inches.*	*Diameter.*	*Length of 1 lb. in inches.*	
	Round Iron	*Square Iron*		*Round Iron*	*Sq*
1/2	19.67	15.38	2 3/16	.86	
5/8	12.63	9.83	2 1/8	.78	
3/4	8.69	6.85	2 5/16	.70	
7/8	6.41	5.02	2 3/4	.64	
1	4.44	3.84	2 7/8	.59	
1 1/8	3.87	3.03	3	.54	
1 1/4	3.13	2.45	3 1/8	.50	
1 3/8	2.58	2.03	3 1/4	.46	
1 1/2	2.17	1.70	3 3/8	.42	
1 5/8	1.85	1.45	3 1/2	.39	
1 3/4	1.59	1.25	3 5/8	.37	
1 7/8	1.39	1.09	3 3/4	.34	
2	1.22	.96	3 7/8	.32	
2 1/8	1.08	.85	4	.30	
2 1/4	.96	.75			

To obtain the length of the weight, multiply the len
pound in inches by the number of pounds which the w
balance.

Window Glass.

nber of panes per 50 feet or in one box.

	Panes in Box.	Size in Inches.	Panes in Box.	Size in Inches.	Panes in Box.	Size in Inches.	Panes in Box.
8	150	12 x 19	32	16 x 20	23	24 x 44	7
9	115	12 x 20	30	16 x 22	20	24 x 50	6
10	90	12 x 21	29	16 x 24	19	24 x 56	5
11	82	12 x 22	27	16 x 30	15	26 x 36	8
12	75	12 x 23	26	16 x 36	12	26 x 40	7
10	80	12 x 24	25	16 x 40	11	26 x 48	6
11	72	13 x 14	40	18 x 20	20	26 x 54	5
12	67	13 x 15	37	18 x 22	18	28 x 34	8
13	62	13 x 16	35	18 x 24	17	28 x 40	6
14	57	13 x 17	33	18 x 26	15	28 x 46	6
15	53	13 x 18	31	18 x 34	12	28 x 50	5
16	50	13 x 19	29	18 x 36	11	30 x 40	6
10	72	13 x 20	28	18 x 40	10	30 x 44	4
12	60	13 x 21	26	18 x 44	9	30 x 48	5
13	55	13 x 22	25	20 x 22	16	30 x 54	5
14	52	13 x 24	23	20 x 24	15	32 x 42	5
15	48	14 x 15	34	20 x 25	14	32 x 44	5
16	45	14 x 16	32	20 x 26	14	32 x 46	5
17	42	14 x 18	29	20 x 28	13	32 x 48	5
18	40	14 x 19	27	20 x 30	12	32 x 50	4
11	59	14 x 20	26	20 x 34	11	32 x 54	4
12	55	14 x 22	23	20 x 36	10	32 x 56	4
13	50	14 x 24	22	20 x 40	9	32 x 60	4
14	47	14 x 28	18	20 x 44	8	34 x 40	5
15	44	14 x 32	16	20 x 50	7	34 x 44	5
16	41	14 x 36	14	22 x 24	14	34 x 46	5
17	39	14 x 40	13	22 x 26	13	34 x 50	4
18	36	15 x 16	30	22 x 28	12	34 x 52	4
12	50	15 x 18	27	22 x 36	9	34 x 56	4
13	46	15 x 20	24	22 x 40	8	36 x 44	5
14	43	15 x 22	22	22 x 50	7	36 x 50	4
15	40	15 x 24	20	24 x 28	11	36 x 56	4
16	38	15 x 30	16	24 x 30	10	36 x 60	3
17	35	15 x 32	15	24 x 32	9	36 x 64	3
18	33	16 x 18	25	24 x 36	8	40 x 60	3

ight of Skylight and Floor Glass per Square Foot.

BRITISH.

ness in inches.....	1/8	3/16	1/4	3/8	1/2	5/8	3/4	1
it in lbs.....	1.75	2.62	3.5	5.35	7.0	8.75	10.5	14.0

AMERICAN.

ness in inches.....	1/8	3/16	1/4	3/8	1/2	5/8	3/4	1
it in lbs.....	1.62	2.43	3.25	4.88	6.5	8.13	9.75	13.0

BUILDING MATERIALS.

Corrugated and Galvanized Iron.

Corrugated iron forms an effective covering for the roofs and sides of buildings. For roofs it is usually laid upon the purlins and secured in place by clips of hoop iron bent around the purlins and riveted to the sheets at about 12 inches apart, taking special care to fasten well the projecting edges of the sheets at the eaves and gable ends of the roof to prevent the wind from loosening and folding them up. The sides of the buildings may be covered directly on the uprights or secured to sheathing. The corrugations vary in size; the smaller giving a more desirable effect, except in the case of very large buildings, where the larger is used, being stiffer and admitting of greater span. Nos. 20 and 22 for both roofing and siding are those more generally used. The result of transverse strain tests on a No. 20 corrugated sheet 2′ 0″ wide, 6′ 0″ long between supports, loaded uniformly with fire clay, was as follows:

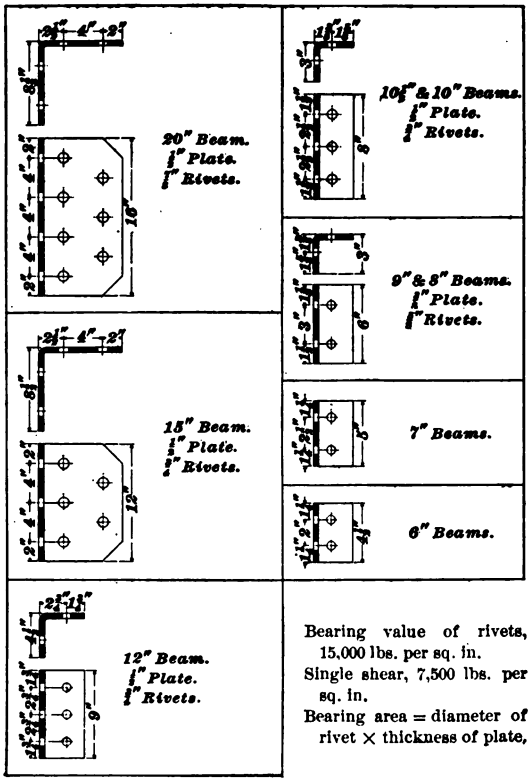
Load per Square Foot. Lbs.	*Deflection at Center under Load. Inches.*	*Permanent Deflection. Load Removed.*
5	1/16	0
10	3/4	0
15	1	0
20	1 1/4	0
25	1 1/2	0
30	1 7/8	1/16
35	2 1/4	1/8
40	2 5/8	3/4
45	3 1/2	1 1/8
50	4	1 1/2
55	6 1/16	Not taken
60	Broke down	Not taken

WEIGHTS OF SHEET IRON, FLAT AND CORRUGATED. CALCULATED FOR FLAT SHEETS 30 1/2 INCHES WIDE.

No. by B'ngham English gauge.	*Thickness.*	*Weight per sq. ft. Flat.*	*Weight per sq. ft. Corrugated.*	*Weight per square of 100 square feet when laid, allowing 6″ lap in length, and 2 1/2″ or one corrugation in width of sheet, for sheet lengths of:*						*Weight per sq. ft. Galvanized.*
				5′	6′	7′	8′	9′	10′	
	in.	*lbs.*	*lbs.*							*lbs.*
16	.065	2.61	3.28	365	358	353	350	348	346	2.95
18	.049	1.97	2.48	275	270	267	264	262	261	2.31
20	.035	1.40	1.76	196	192	190	188	186	185	1.74
22	.028	1.12	1.41	156	154	152	150	149	148	1.46
24	.022	.88	1.11	123	121	119	118	117	117	1.22
26	.018	.72	.91	101	99	97	97	96	95	1.06

Standard Beam Connections.

U. S. GOVERNMENT PRACTICE.



Bearing value of rivets, 15,000 lbs. per sq. in.

Single shear, 7,500 lbs. per sq. in.

Bearing area = diameter of rivet × thickness of plate,

Safe Loads in Tons of 2,000 lbs. for Hollow Cylindrical Cast Iron Columns.

$\frac{1}{10}$th of the Ultimate Strength.

Outside diam. in inches.	Thickness of metal.	Length of Columns, in feet.									Sectional Area in inches.	Weight in lbs. of Columns per foot of length.
		8	10	12	14	16	18	20	22	24		
		Tons.	*Tons.*	*Tons.*	*Tons.*	*Tons.*	*Tons.*	*Tons.*	*Tons.*	*Tons.*		
6	¾	37.5	33.0	28.8	25.0	21.7	18.9	16.5	....	....	12.4	38.59
6	1	47.6	41.9	36.5	31.8	27.6	24.0	21.0	....	....	15.7	49.01
7	¾	47.7	43.1	38.5	34.3	30.4	26.9	23.9	21.2	18.9	14.7	45.96
7	1	61.1	55.2	49.3	43.8	38.9	34.4	30.6	27.1	24.2	18.9	58.90
8	¾	57.9	53.3	48.6	44.1	39.7	35.8	32.2	28.9	26.1	17.1	53.29
8	1	74.6	68.7	62.5	56.7	51.1	46.0	41.4	37.3	33.6	22.0	68.64
9	¾	68.1	63.6	58.9	54.2	49.6	45.2	41.2	37.5	34.1	19.4	60.65
9	1	88.0	82.3	76.2	70.0	64.1	58.4	53.2	48.4	44.1	25.1	78.40
9	1¼	106.6	99.6	92.2	84.8	77.6	70.8	64.4	58.7	53.4	30.4	94.94
10	1	101.4	95.9	89.8	83.6	77.4	71.5	65.8	60.5	55.5	28.3	88.23
10	1⅛	143.7	135.8	127.3	118.5	109.7	101.2	93.2	85.6	78.7	40.1	124.99
10	1¼	162.7	153.8	144.1	134.1	124.2	114.6	105.5	97.0	89.1	45.4	141.65
11	1	114.8	109.4	103.5	79.3	91.0	84.8	80.2	73.1	67.7	31.4	98.03
11	1¼	139.9	133.3	126.1	118.6	110.9	103.3	97.8	89.4	82.5	38.3	119.46
11	1⅜	185.7	177.1	167.5	157.5	147.3	137.2	129.8	118.3	109.5	50.9	158.68
12	1	128.0	122.9	117.2	111.0	104.7	98.4	92.2	86.1	80.4	34.6	107.51
12	1¼	156.4	150.1	143.1	135.7	127.9	120.2	112.6	105.2	98.2	42.2	131.41
12	1½	183.3	175.9	167.7	159.0	149.9	140.9	132.0	123.3	115.1	49.5	154.10
12	2	232.7	223.4	213.0	201.9	190.4	178.9	167.6	156.6	146.1	62.8	195.75
13	1	141.2	136.3	130.7	124.7	118.5	112.1	105.8	99.5	93.5	37.7	117.53
13	1¼	172.8	166.8	160.0	152.7	145.0	137.2	129.4	121.8	114.4	46.1	143.86
13	2	258.9	249.9	239.7	228.7	217.3	205.5	193.9	182.5	171.3	69.1	215.56
14	1	154.3	149.0	144.3	138.5	132.3	125.9	119.5	113.1	106.8	40.8	127.60

TABLE—*Continued.*

Outside diam. in inches.	Thickness of metal.	Length of Columns, in feet. 8	10	12	14	16	18	20	22	24	Sectional area in inches.	Weight in lbs. of columns per foot of length.
		Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.	Tons.		
14	1¼	189.2	183.4	176.9	169.7	162.2	154.4	146.5	138.6	131.0	50.1	156.31
14	1⅜	222.6	215.8	208.1	199.7	190.8	181.7	172.3	163.1	154.1	58.9	183.67
14	1½	284.8	276.2	266.4	255.6	244.2	232.4	220.6	208.8	197.2	75.4	235.12
15	1	167.4	162.9	157.8	152.1	146.0	139.7	133.3	126.8	120.4	44.0	137.28
15	1¼	205.5	200.0	193.7	186.7	179.3	171.5	163.6	155.7	147.9	54.0	168.48
15	2	310.8	302.5	293.0	282.5	271.2	259.5	247.5	235.5	223.6	81.7	254.90

Safe load on standard wrought iron pipe used as columns.
In tons of 2,000 lbs. Both ends fixed. Factor of safety = 6.

Size of Pipe.	Length of Column. 8 ft.	10 ft.	12 ft.	14 ft.	16 ft.	18 ft.	20 ft.
2 in.	2.0	1.8	1.4	....	....	....	....
2½ "	....	3.35	2.8	2.4	2.1	....	....
3 "	....	4.8	4.3	3.8	3.36	3.08	....
3½ "	....	6.07	5.52	5.1	4.47	4.02	....
4 "	....	7.67	7.1	6.56	6.02	5.4	....
4½ "	....	9.32	8.69	8.16	7.52	6.98	....
5 "	....	...	10.53	9.93	9.33	8.6	8.0
6 "	....	....	14.6	13.82	13.03	12.37	11.75
7 "	....	....	18.58	17.9	17.2	16.5	15.76
8 "	....	....	23.13	22.45	21.7	20.85	20.11

NUMBER OF NAILS AND TACKS PER POUND.

Nails.			*Tacks.*		
Title.	*Length in inches.*	*No. nails per lb.*	*Title.*	*Length in inches.*	*No. tacks per lb.*
3 Penny fine	1½	760	1 oz.	½	16,000
3 " "	1¼	480	1½ "	⅝	10,666
4 " "	1½	300	2 "	¾	8,000
5 " "	1¾	200	2½ "	⅞	6,400
6 " "	2	160	3 "	1	5,333
7 " "	2¼	128	4 "	1¼	4,000
8 " "	2½	92	6 "	1½	2,666
9 " "	2¾	72	8 "	1¾	2,000
10 " "	3	60	10 "	2	1,600
12 " "	3¼	44	12 "	2¼	1,333
16 " "	3½	32	14 "	2½	1,143
20 " "	4	24	16 "	2⅝	1,000
30 " "	4½	16	18 "	2¾	888
40 " "	5	14	20 "	3	800
50 " "	5½	12	22 "	1 1/4	727
60 " "	6	10	24 "	1⅛	666
6 " fence	2	80			
8 " "	2½	50			
10 " "	3	34			
12 " "	3¼	29			

NATIONAL WIRE NAIL CARD

ADOPTED DEC. 1, 1896,

ON WHICH PRICES OF WIRE AND CUT NAILS ARE BASED.

Common, Fence, Shingle, Tobacco, Flooring, and Common Brads.

	ADVANCE
20d to 60d	Base
10d to 16d	$0.05
8d and 9d	.10
6d and 7d	.20
4d and 5d	.30
3d	.45
2d	.70

Barbed Common and Barbed Car Nails.

15 cts. advance over common.

Casing and Smooth Box.

10d and larger	$0.15
8d and 9d	.25
6d and 7d	.35
4d and 5d	.50
3d	.70
2d	1.00

Barbed Box 15 cents advance over smooth.

NATIONAL WIRE NAIL CARD—*Continued.*

Smooth Finishing Nails.

	ADVANCE
10d and larger	$0.25
8d and 9d	.35
6d and 7d	.45
4d and 5d	.65
3d	.85
2d	1.15

Fine Nails.

2d	$1.00
3d	.50

Lining Nails.

¾ inch	$1.20
⅞ inch	1.00
1 inch	.80

Slating Nails.

2d	$0.80
3d	.60
4d and 5d	.40
6d	.30

Barrel Nails.

¾ inch	$1.00
⅞ inch	.85
1 inch	.70
1⅛ inch	.60
1¼ inch	.50
1⅜ inch	.40
1½ inch	.30

Barbed Roofing Nails.

	ADVANCE
¾ inch	$0.75
⅞ inch	.65
1 inch	.60
1⅛ inch	.60
1¼ inch	.55
1½ inch	.45
1¾ inch	.45
2 inch	.35

Clinch Nails.

2d	$1.05
3d	.85
4d and 5d	.65
6d and 7d	.55
8d and 9d	.45
10d to 20d	.35

Hinge Nails.

4d	$0.80
6d	.70
8d	.60
10d and larger	.50

Spikes.

All sizes	$0.10

Barbed Dowel Pins.

⅝ inch	$2.00
¾ inch	1.75
⅞ inch	1.50
1 inch	1.25
1⅛ inch	1.15
1¼ inch	1.00
1⅜ inch	1.00
1½ inch	1.00

The table above gives price per keg of all varieties of nails—the rates are computed on the price of a keg of from 20 to 60 penny as a *base*—to which must be added the advance rate for the smaller sizes as shown.

The *wholesale* base for wire nails is $1.70 and for cut nails, $1.60.

EXAMPLE.—To determine price of a keg of 8 penny wire nails: To the base of $1.70 add the advance rate of 10c., giving $1.80 the price per keg.

Cut Nails vs. Wire Nails.

A series of interesting experiments was recently made at the U. S. Arsenal at Watertown to decide upon the relative merits of cut and wire nails.

At the first test the size of the cut nails ranged from 1½' nails to 6" spike, forty penny and sixty penny, and ranging in number per pound from six to seventeen. The wire nails were of approximately the same size and weight.

The nails of the respective sizes were driven into a spruce plank to a uniform depth—the larger spike to four inches.

The total weight of the nails differed but two grammes, the wire nails weighing 214 and the cut nails, 212 grammes.

In the first test a force of 733 pounds was required to draw the wire nail and 836 pounds for the cut nail. The second wire nail was drawn with force of 673 pounds, while 742 pounds were required for the cut nail. The third wire nail required 675 and cut nail, 864 pounds. The fourth wire nail, 594 and the cut nail, 964 pounds. The fifth and sixth nails were drawn at nearly the same ratio. The seventh wire nail was drawn at 879 pounds, while it required 1,200 pounds for the cut nail.

Care was taken to have the tests fairly made and accurate. The results thus obtained would indicate that cut nails are superior to wire nails for work when dependence is placed on nails to hold it together.

Slating.

Slating is calculated by the "Square," meaning the area of 100 square feet, and is usually laid in 3 thicknesses, that is, the third laps over the first three inches, with the second one between. To ascertain the number of slates of a given size, required per square, substract 3 inches from the length of the slate, multiply the remainder by the width and divide by 2 (this will give the number of square inches covered per slate); then divide 14,400 (the number of square inches contained in a square) by the number found as above, and the result will be the number

of slates required to cover a square. The following table gives the number of slates per square for the usual stock sizes:

Size in Inches.	*Pieces per Square.*	*Size in Inches.*	*Pieces per Square.*	*Size in Inches.*	*Pieces per Square.*
6 × 12	533	8 × 16	277	12 × 20	141
7 × 12	457	9 × 16	246	14 × 20	121
8 × 12	400	10 × 16	221	11 × 22	137
9 × 12	355	9 × 18	213	12 × 22	126
7 × 14	374	10 × 18	192	14 × 22	108
8 × 14	327	12 × 18	160	12 × 24	114
9 × 14	291	10 × 20	169	14 × 24	98
10 × 14	261	11 × 20	154	16 × 24	86

The weight of slate per cubic foot is about 175 lbs., or, per square foot of the usual thicknesses, as follows:

Thickness in inches.....	1/8	3/16	1/4	5/16	1/2
Weight in lbs. per sq. ft.....	1.82	2.73	3.64	5.46	7 28

The weight of slating will depend on the size used.

For example a 10 × 18 slate = 1.25 sq. ft. of surface; if 3/16 in. thick, its weight will be 1.25 × 2.73 = 3.41 lbs., and as there are 192 slates in a square, the weight of a square will be 192 × 3.41 = 654 lbs. or 6.54 lbs. per square foot.

Scaffolding with Round Spars.

In erecting a scaffolding by this method, the uprights and cross-pieces supporting the platform do not require to be nailed or bolted together. The effect of the wind on a nailed structure is to loosen the joints and thus reduce its strength. Rain and dampness will cause the wood to swell, which, afterward shrinking, will cause loosening and splitting at the joints.

Nailed scaffolding is always more or less injured and broken during removal, after a job is completed, and can seldom be used more

than two or three times. Lashed scaffolding, on the contrary, may oscillate with the wind without loosening the joints; it is unaffected by extremes of temperature further than the usual deterioration of ropes and timber, and can be repeatedly used

The upright poles should be proportioned to suit the height of the scaffold and such load as may be placed upon it. Spruce poles from which the bark has been stripped, measuring from 6" to 7" diameter at the large end, and from 2" to 3" at the small

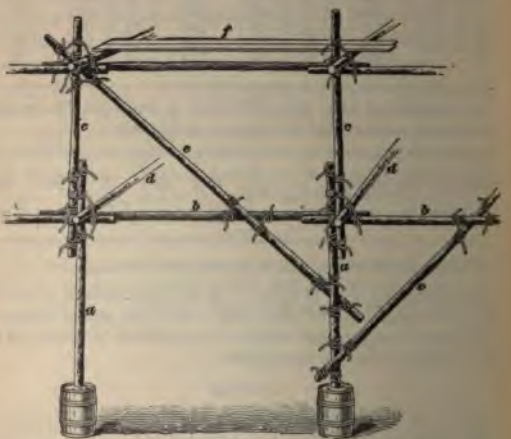


FIG. 1.

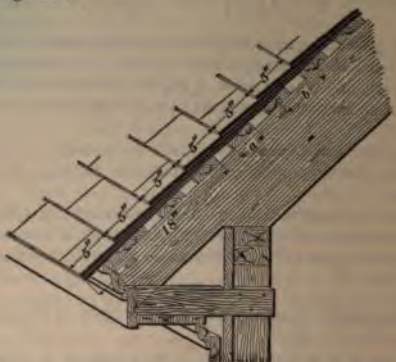
end and 20 feet in length will suit under ordinary conditions; the same sized timber can be used for the horizontal and diagonal spars. In tying or lashing the members together, the loose knot called the "clove hitch" is generally used. It possesses several advantages over other forms of knots in that it will not untie or loosen, never slips, and can be readily removed.

In Fig. 1 are shown two barrels filled with earth or sand in which the upright poles *a*, *a* are placed. Near the upper ends of the uprights and at right angles to them, are lashed the longitudinal ties or ledgers *b*, *b*. Extension poles, *c*, *c*, are lashed to the sides of the uprights *a*, *a*. Cross-ties or *putlogs* *d*, *d*, rest on the ledgers, and are lashed both to the ledgers and the uprights, the opposite ends resting on the walls of the building. Diagonal spars or braces *e*, *e*, are also lashed to both the ledgers and uprights, as shown, to keep the scaffolding from swaying. Planking *f*, from 2 to 3 in. thick is laid on the putlogs, and forms the platform for the workmen, and such material as they may require.

FIG. 2.

Fig. 2 shows the method of lashing the several pieces together, the clove hitch being used for all conditions. The mode of forming the hitch may thus be described: Beginning at *c*, the rope is wound around the pole *a*, from *c* to *d*; thence around the back of the pole, emerging at *e*; thence in a loose loop to *f*, around the pole to *g*, and under the loop to *h*, around to *i*, and again passing under the loop, emerging at *j*; then passing around and hitching to pole *b* in a similar manner.

the lap *a*, of the fourth shingle over the first is 3". Having decided on lap, which is usually from 2" to 3", subtract it from the length of the shingle and divide the difference by 3, which will give the exposed length. Thus with a shingle 18" long and the lap 3", the exposed length will be 18" — 3" = 15", which divided by 3 will give 5".



Shingles are preferably placed on slats from 1" to 1¼" thick and 2" to 3" wide, the distance *b*, between centers of slats being equal to the exposed length of the shingle.

The following table is arranged for shingles from 16" to 27" in length, and is based on a given exposure to the weather:

Exposure to the Weather.	*No. of sq. ft. of Roof covered by 1,000 Shingles.*		*No. of Shingles required for 100 sq. ft. of Roof.*	
	4" wide.	*6" wide.*	*4" wide.*	*6" wide.*
4 Inches.	111	167	900	600
5 "	139	208	720	480
6 "	167	250	600	400
7 "	194	291	514	343
8 "	222	333	450	300

The number of inches exposed to the weather multiplied by the average width of a shingle will give the area of the exposure; dividing 14,400, the number of square inches in 100 square feet, by the area of the exposure will give the number of shingles required to cover 100 square feet of roof.

NOTES RELATIVE TO CLOCK AND BELL TOWERS.

The *dial* of the clock, to show plainly, should be 1 foot in diameter for every 10 feet of height above the ground, and this ratio should be increased when it is intended that it should be clearly seen at a distance. The dial should never be made convex, owing to the difficulty of placing the hands so that they can be in close contact with the face; if the minute hand is bent to avoid the difficulty, it presents an unsatisfactory appearance. The dial should be concave in form, and may be of stone, slate or copper, having the middle portion countersunk for the hour hand, so that the minute hand may go close to the figures and avoid parallax.

Where the dials are large, they can be formed directly in the masonry by having the surface clean cut, as most kinds of stone hold the paint and gilding better than any of the metals do. The dial should not be sunk into the face of the wall like a panel, as it will not receive sufficient light during the day, and being sunk, its face cannot be thoroughly washed by the rain.

A rule which has been adopted after many experiments, as the best for proportioning the figures on the dial, may be thus stated. Divide the radius into three, and leave the inner two-thirds clear and flat, and of some color to form a strong contrast with the color of the hands, black or deep blue if they are gilt, and white if they are black and *vice versa*. The figures should occupy the next two-thirds of the remaining third, with the minutes set in the remainder, near the edge, having every fifth minute more strongly marked than the rest.

The rim of the dial (if any), and the counterpoise at the center, should never be painted or gilded to match the hands as it causes optical confusion.

The long hand should be straight and plain, to distinguish it from the minute hand which should have a "heart" or swell

near the index-point and should be counterpoised externally in order to withstand the force of the wind.

Metallic *skeleton dials* are to be avoided, as there is no proper background to form a contrast with the hands.

Where *illuminated dials* are used, they are generally made of white opaque glass with metallic figures and a rim, the lighting being accomplished with gas or electric light.

Proper provision must be made for the installation of the clock and its weights.

The *Belfry* should be at least 10 feet above the roof of the building, and, preferably, should not be much larger than to conveniently contain the bell.

The *windows* should reach from the bell deck to the ceiling, the *slats* should be from 8 to 10 inches apart, not more than 10 to 12 inches wide, and at an angle of not more than 20° from the horizontal.

The *floor* should be water-tight, and as near the window sills as possible.

The *ceiling* should be tightly closed in all cases and placed within 1 foot of the bell when in position.

The *mouth of the bell* should be from 2½ to 3½ feet above the window sills.

In order to prevent the passage of birds through the slats, it is well to cover them with No. 18 B. W. G. ½" mesh wire cloth, as many fires have occurred through accumulations of twigs and straw used by the birds in building their nests in towers.

PAINTERS' MEMORANDA.

1 gal.	priming color	will cover	50	sq. yds.
1 "	white zinc	"	50	"
1 "	" paint	"	44	"
1 "	" lead color	"	50	"
1 "	" black paint	"	50	"
1 "	" stone color	"	44	"
1 "	" yellow paint	"	44	"
1 "	" blue color	"	45	"
1 "	" green paint	"	45	"
1 "	" bright emerald green	"	25	"
1 "	" bronze green	"	75	"

One pound of paint will cover about 4 sq. yds. the first coat, and about 6 sq. yds. each additional coat. One pound of putty is required for stopping about every 20 sq. yds. A mixture of 1 gal. of tar and 1 lb. of pitch will cover about 12 sq. yds., the first coat, and about 17 sq. yds. each additional coat. A sq. yd. of new brick wall requires for the first coat of paint in oil about ¾ lb.; for the second, about 3 lbs.; and for the third, about 4 lbs.

PIPES AND FITTINGS.

Cast iron soil pipes should be uniform in thickness and homogeneous throughout. They should be tested with water pressure and then coated with asphaltum before being used. Any pipes lighter than the following should be rejected :

Nominal Diameter.	*Weight Per Linear Foot.*
2 inches	5½ pounds.
3 "	9½ "
4 "	13 "
5 "	17 "
6 "	20 "
7 "	27 "
8 "	33½ "
10 "	45 "
12 "	54 "

These pipes come in 5-ft. lengths and are known as *extra heavy*. The maker's name should preferably be cast on each pipe and fitting.

Cast iron soil pipe fittings should correspond with the grade of pipes used. They should have easy curves. No sharp 90° bends or T branches should be used ; only obtuse angle fittings. Fittings are made as staple goods at the following angles : 90°, 45°, 22½°, 11¼°, and are known as quarter, eighth, sixteenth and thirty-second fittings, respectively.

Cast iron soil pipe joints are made with picked oakum and *molten lead caked* solidly home in the sockets. Twelve ox-

of soft pig lead must be used in each joint for each inch in diameter of the pipe.

Wrought iron and steel pipes for drainage purposes should be stamped with the maker's name. They should be galvanized and must conform to the following table:

Nominal Diameter.	*Thickness of Metal.*	*Weight Per Linear Foot.*
1½ inches.	.14 inches.	2.68 pounds.
2 "	.15 "	3.61 "
2½ "	.20 "	5.74 "
3 "	.21 "	7.54 "
3½ "	.22 "	9.00 "
4 "	.23 "	10.66 "
4½ "	.24 "	12.34 "
5 "	.25 "	14.50 "
6 "	.28 "	18.76 "
7 "	.30 "	23.27 "
8 "	.32 "	28.18 "
9 "	.34 "	33.70 "
10 "	.36 "	40.06 "
11 "	.37 "	45.02 "
12 "	.37 "	48.98 "

Fittings for *vent pipes* on wrought iron or steel pipes may be the ordinary cast or malleable steam and water fittings.

Fittings for *waste* or *soil pipes* must be the special, extra-heavy cast iron recessed and threaded drainage fittings, with smooth interior water-way and threads tapped, so as to give a uniform grade to branches of not less than ¼ of an inch per foot.

All joints must be screwed joints made up with red lead, and the burr formed in cutting must be carefully reamed out.

Short nipples on wrought iron or steel pipe where the unthreaded pipe is less than one and one-half inches long must be of the thickness and weight known as "extra heavy" or "extra strong."

Brass soil, waste and vent pipes, and solder nipples should be thoroughly annealed seamless drawn brass tubing of standard Iron pipe gauge. Connections on brass pipe and between brass

pipe and traps or iron pipe must not be made with slip joints or couplings. Threaded connections on brass pipe should be tapered and of the same size as iron pipe threads.

The following average thicknesses and weights per linear foot should be employed :

Nominal Diameter.	*Thickness.*	*Weight Per Linear Foot.*
1¼ inches.....	.14 inches.	2.84 pounds.
2 "	.15 "	3.82 "
2½ "	.20 "	6.66 "
3 "	.21 "	7.92 "
3½ "	.22 "	9.54 "
4 "	.23 "	11.29 "
4½ "	.24 "	13.06 "
5 "	.25 "	15.37 "
6 "	.28 "	19.88 "

Brass ferrules should be bell-shaped, extra heavy cast brass, not less than four inches long and two and one-quarter inches in diameter, and not less than the following weights :

Inside Diameter.	*Weight, Each.*
2¼ inches.....	1 pound 0 ounces.
3½ "	1 " 12 "
4½ "	2 " 8 "

Cast brass solder nipples should be of the following weights :

Inside Diameter.	*Weight, Each.*
1¼ inches	0 pound 8 ounces.
2 "	0 " 14 "
2½ "	1 " 6 "
3 "	2 " 0 "
4 "	3 " 8 "

Lead soil, waste and vent pipes should be of the following weights :

Nominal Diameter.	*Weight Per Linear Foot.*
1¼ inch (for flush pipes only)	2½ pounds.
1½ inches	3 "
2 "	4 "
3 "	6 "
4 and 4½ inches	8 "

All lead traps and bends should be of the same weights and thicknesses as their corresponding pipe branches. This grade is known in commerce as **D**.

Sizes of Pipes.

The size of the house sewer and house drain must be at least four inches in diameter where water-closets discharge into them.

Where rain water discharges into them, the house sewer and the house drain up to the leader connections should be in accordance with the following table :

Diameter.	*Fall ¼-in. Per Foot.*	*Fall ⅛-in. Per Foot.*
6 inches	5,000 sq. ft.	7,500 sq. ft. drainage area.
7 "	6,900 "	10,300 " "
8 "	9,100 "	13,600 " "
9 "	11,600 "	17,400 " "

The sizes of soil and waste pipes should be *not less* than those given in the following table :

Name of Pipe.	*Diameter in Inches.*
Main soil pipe	4
" waste pipe	2
Branch soil "	4
" waste pipes for kitchen sinks	2
Soil pipe for water-closets on five or more floors	5
Waste pipe for kitchen sinks on five or more floors	3

SIZES OF WASTE BRANCHES FOR FIXTURES.

Name of Pipe.	*Diameter in Inches.*
Bath waste pipe	1½ to 2
Basin " "	1¼ " 1½
Urinal " "	1½ " 1½
Sink " "	1½ " 2
Pantry sink waste pipe	1½
Safe waste pipe	1 " 1½
Water-closet trap	3¼ " 4
Wash tubs, 1¼" branch and 2" trap for set of two tubs	1½ " 2
Set of three or four tubs	2

The sizes of vent pipes may be found by the following table:

Name of Pipe.	*Diameter in Inches.*
Main vents and long branches	2
Water-closet vents on three or more floors	3
Other fixtures on less than seven floors	2
Fixtures on eight stories or less	3
Eight stories and less than sixteen	4
Fifteen " " " " twenty-two	5
Twenty-one stories and over	6
Branch vents for traps larger than 2"	2
" " " " 2" or less	1½

For fixtures other than water-closets and slop sinks and for more than eight stories, vent pipes may be one inch smaller than above stated.

All vent pipes which pass out through the roof should be increased one size, to allow for ice accumulations inside. In no case should any of these pipes be less than 4" in diameter, except in mild climates.

The *fresh air inlet* should be of the same size as the drain, up to four inches; for five-inch and six-inch drains it should be not less than four inches in diameter; for seven-inch and eight-inch drains not less than six inches in diameter, and for larger drains not less than eight inches in diameter.

The fresh air inlet orifice should have an area equal to that of the pipe, and should be set at least 15 feet away from any door or window.

Proper Fall for Drains and Waste Pipes.

The *fall* or *pitch* of all parts of a drainage system should be so arranged that the velocity of the flow obtained will be not less than about 275 feet per minute. This velocity can be closely approximated by pitching the pipes as follows:

Diameter of Pipe in Inches.	*Proper Fall.*
2	1 foot fall in 20 foot run.
3	1 " " 30 "
4	1 " " 40 "
5	1 " " 50 "
6	1 " " 60 "
7	1 " " 70 "
8	1 " " 80 "
9	1 " " 90 "
10	1 " " 100 "

SANITARY MAXIMS.

(1) General water-closet accommodation of tenements should never be placed in cellar or basement.

(2) All water-closet apartments should be located where plenty of daylight and ventilation can be obtained.

(3) All water-closet apartments must open to the outer atmosphere either direct, or by air shafts at least 3 feet square.

(4) To prevent damp or wet cellars, subsoil drainage should be employed where necessary. They must be effectively trapped from sewers or house drains, and some means must be employed to maintain a seal. A check valve, or back-water trap should be used to prevent a back discharge of sewage into them, should the drains become choked.

(5) The arrangement of all drainage or vent pipes should be as direct as possible.

(6) If there is a sewer in a street, every building should connect to it separately.

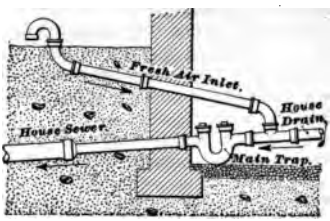
(7) Where the soil is "natural," the house sewer or sewer connection may be of vitrified earthenware pipe, uniformly bedded *and jointed* with Portland cement and clean sharp sand. This *pipe must* run straight; and must have a clear bore.

(8) If the soil is filled in or "made," the house sewer must be of extra heavy cast iron, asphalt coated, wrought iron, galvanized and asphalt coated, or brass pipes, to avoid leakage by a settlement of the earth.

(9) Where there is no sewer in the street, and when it is necessary to run a private sewer to connect with a sewer in another street, it should be laid outside of the curb and under the street on which the buildings face, not in yards or across lots where buildings may in future be erected.

(10) The main house drains should be run above the cellar floor when possible, and be secured against the cellar walls, supported upon piers built under each joint, or properly suspended from the cellar ceiling by adjustable hangers.

(11) If they must be run under the cellar floors they should be laid in perfectly straight runs, and clean-outs or inspection fittings should be placed at each branch or change in direction of the runs.



(12) All changes in direction should be made with curved pipes or Y branches and $\frac{1}{16}$ or $\frac{1}{8}$ bends.

(13) Old sewers should never be employed for new buildings unless first examined and tested by the smoke machine.

(14) All house drainage systems should be disconnected from city sewers by means of a main disconnecting trap. (See figure.)

If the sewage from a country building delivers into the sea, a river or openly (into a creek, for instance) the main drain trap may, in many cases, be advantageously omitted. A fresh air inlet must in all cases accompany a main drain trap.

(15) Fresh air inlet orifices must be at least 15 feet from the nearest window or door, and no cold air box for a furnace should be so placed as to draw air from them.

(16) All vent outlets should discharge at least 2 feet above the highest part of a roof ridge, coping or light shaft, and as far as possible away from the light shaft and all water tanks.

(17) Vent pipes above roofs must be 4" diameter or larger, and no cowls or vent caps should be used.

(18) Vent pipes passing up through a low roof and within 30 feet of any windows in taller adjoining buildings should be extended to safe points above the higher roofs.

(19) All drain, soil, waste and vent pipes between the main disconnecting trap and the vent outlets must be clear and unobstructed by traps, check-valves or other contrivances.

(20) All house drainage work should be as accessible as possible, being placed either on the faces of walls or in air shafts. When necessarily placed in walls they should be covered by movable pipe boards.

(21) Every fixture in a building must be separately trapped close to each fixture, except where a sink and wash tubs are side by side and immediately adjoin each other, in which case the waste pipe from the tubs may join the inlet side of the sink trap and below the water seal.

(22) All fixture and other such traps in a building must be back-vented by a separate pipe which may deliver into a special back-vent stack at a point about 2 or 3 inches below top of fixture, for tenement or apartment buildings, and at much higher points if desired for private residence work, but in no cases at points lower than bottoms of fixtures or bowls.

(23) Where lead waste or back-vent branches connect to cast iron stacks, the connections must be made with heavy brass ferrules and wiped solder joints; if to wrought iron or brass stacks, by means of brass-screwed solder nipples provided with a socket to receive the lead pipe and form a flush internal surface. All solder joints in a plumbing system should be "wiped."

(24) Special precaution should be taken to secure perfect joints between water-closet traps placed above the floor and the branch soil and vent pipes for same. Brass floor plates should be used for the floor connections. A smoke test is necessary to prove these joints. Back-vent horn or porcelain traps should not be permitted, as they soon break off. The best modern

practice is to back-vent the soil pipe waste close to the floor connection by means of a wiped or screwed joint.

(25) Overflow pipes from all fixtures must connect to traps on house side of their seals.

(26) All fixture safes should be properly graded to a special waste pipe which must deliver openly at some point, such as a safe-waste sink in the cellar. The outlets of these pipes should be covered with light-flap valves.

(27) The sediment pipe from the kitchen boilers should not be connected on the outlet side of the sink or other trap.

(28) A separate small tank or cistern should be employed to flush every water-closet, and in no case should any water or urinal closet be supplied directly from the street pressure pipes when there is any liability of a draw-back in the street mains.

(29) One water-closet, at least, should be allowed for fifteen inmates of a building. Every story of a tenement building should contain at least one water-closet.

(30) Drinking water should be drawn directly from the street mains. Tank water may be used for washing and bathing purposes.

(31) House tanks should be made of wood, and if placed inside the building, should be lined with tinned and planished sheet copper.

(32) Outside tanks may be made circular of cedar staves or wrought iron. Overflow from such tanks should discharge on the roof.

(33) Overflow pipes from inside tanks may deliver into an eaves gutter if one is available, or they may be trapped and discharged into an open sink.

No pipe connecting to a tank should deliver directly into the drainage system.

(34) All rain-water leaders, area-water boxes and subsoil drainage, must be trapped from the drainage system, and the seals maintained.

(35) Steam-exhaust, blow-off or drip-pipes should not deliver directly into a drainage system. Such water should deliver through a deep seal trap, and at a temperature not higher than 100° F.

No privy-vaults or cesspools for sewage should be permitted in any place where water-closets can be connected with a city sewer.

(37) All architects should see to it that every drainage system when completed, is tested with smoke under a pressure of at least 1" water column, because the sanitary arrangements are seldom perfect when this final test is neglected.

INSPECTION AND TESTING OF DRAINAGE SYSTEMS.

Drainage systems in all respectable cities are inspected and tested twice before they are passed by the authorities as perfectly sanitary. These tests are: 1. A test of the "roughing in," which consists of the iron or brass drains, soil-waste and vent lines and sometimes the fixture branches, also, before any pipes are concealed. 2. A test of the entire system when the fixtures are all set, the traps sealed with water, and the work otherwise complete.

The first or roughing test is accomplished by closing all branches and filling the entire system with water (if the weather permits), allowing the water to stand in the pipes for a certain length of time, which will depend upon the inspector's judgment. If any leaks exist they can be detected by water flowing from them. In applying this test, particular care must be taken to use plugs in the lower openings, which cannot be blown out by the heavy pressures that occur at these points.

Should the weather be too cold to apply a water test, the compressed air test is applied. In this case air is pumped into the drainage system until the pressure is 10 lb. by the gauge, when a valve between the pump and the system is closed and the gauge allowed to indicate the air pressure.

By this test the presence of leaks is made manifest by the gauge indicating a decreasing pressure as the test is allowed to remain on the system. The exact locality of the leak, however, is too difficult to determine unless some pungent volatile oil, such as ether or oil of peppermint, is allowed to vaporize within the system and thus cause an odor in the immediate vicinity of the *leaks*.

The second or final test is really the more important test of the two. The chief object is to positively ascertain (a) if the entire system when completed is perfectly gas-tight; (b) if the traps have perfect seals; (c) if every part of the system is trapped that should be trapped; (d) if any back-vent pipes are run into hollow partitions, attics or chimneys. This test was formerly made by vaporizing oil of peppermint in the drainage system without pressure, trusting to a diffusion of a pungent vapor to indicate any leaks.

This form of peppermint test is now abandoned by modern sanitary engineers as untrustworthy, and the smoke test is being universally employed instead. This is applied by blowing smoke through the system; and when the smoke shows at the various vent outlets, they are closed tight and the drains are subjected to a smoke pressure of 1" to 1¼" of water column.

This pressure is sufficient to force smoke through the most minute leaks, and the pressure is not sufficient to blow through the seal of a good trap.

House drainage systems should be tested once a year, and a report of the sanitary condition should be furnished after each inspection and test.

The best smoke machine which can be employed for testing house drains is that composed of a double-action blower combined with a smoke-generating chamber which is furnished with a balanced floating cover. Such a blower will force air through the fire in a steady stream and a uniform efflux of dense smoke is obtained.

The advantage of the floating cover is that it will rise in its water seal when the desired pressure is raised in the drainage system, and the pressure cannot be increased sufficiently to force the seals of traps in the building, because the excess of smoke will escape to the atmosphere from under the cover.

The smoke machine may be applied to the fresh air inlet; or to one of the vent pipes above the roof, preferably to the latter, because any smoke which escapes while lighting the fuel, which is *oily cotton waste*, cannot enter the building and thereby spoil *the test.*

WATER SUPPLY AND DISTRIBUTION.

SOURCES OF SUPPLY.

The source of supply of water to a building will depend upon prevailing conditions and the location of the building. City buildings are in nearly all cases supplied from city mains, while country buildings are to a large extent supplied from wells, lakes and streams.

Street service pipes should connect to the street mains by "corporation stops." A stop and waste should in all cases be placed under the sidewalk at the curb. A separate stop and waste should in all cases be placed upon the service pipe just inside the cellar well.

If the street pressure is great enough to cause water to rise to the top floor of a building, all fixtures in the building are usually applied direct from the mains with both hot and cold water.

If the pressure is too low or the supply intermittent, a tank is placed in the attic to supply the building.

If the pressure is too high, it is customary to apply a pressure regulator in the cellar.

The sizes of street service pipes depend chiefly upon the street pressure and the size of building to be supplied. The following is common practice :

Single dwellings, two or three stories high..	½" or ¾" diameter
Larger dwellings.....	1" " 1¼" "
Tenement buildings and apartment houses.	1½" " 2" "
Hotels and factories.....	2" and up "

These pipes should be increased one size if the pressure is low.

Pumps.

The amount of water raised by single-acting pumps is commonly estimated by multiplying the number of strokes which the piston travels in one direction per minute by the volume displaced or traversed by the piston in a single stroke, the supposition being that the water flows into the pump barrel only when the piston ascends. It has been found, however, that the *column of* water does not cease flowing when the piston descends, *and that* the amount of water delivered is greater than usually

supposed; in some cases, it is nearly double the theoretical amount.

A column of water 34 ft. high balances the pressure of the atmosphere, but in practice it requires a very good pump to *draw* to a height of 28 feet.

Pumping Hot Water. The height to which hot water can be raised by *suction* is much less than that of cold water, the height varying with the temperature. This is due to the increased pressure of the vapor. Where possible, the hot water should flow into the pump by gravity.

The following table gives the maximum vertical heights of suction pipes for different temperatures:

Absolute Pressure of Vapor in lb. per Sq. In.	*Vacuum. Inches of Mercury.*	*Temperature. Degrees, Fahr.*	*Maximum depth of Suction. Feet.*
1	27.88	101.4	31.6
2	25.85	126.2	29.3
3	23.81	144.7	27.0
4	21.77	153.3	24.7
5	19.74	162.5	22.4
6	17.70	170.3	20.1
7	15.66	177.0	17.8
8	13.63	183.0	15.5
9	11 59	188.4	13.2
10	9.55	193.2	10.9
11	7.51	197.6	8.5
12	5.48	201.9	6.2
13	3.44	205.8	3.9
14	1.40	209.6	1.6

The Hydraulic Ram.

This machine is employed to raise water to a point higher than the source of its supply.

It is chiefly used where a large flow of water with a low fall is obtainable, and it raises part of the water which operates it. The *efficiency* of the hydraulic ram varies with the ratio of the fall of the drive pipe with the rise of the discharge, about as follows:

Ratio of lift to fall, 4. 6, 8, 10, 12, 14, 16, 18, 20, 22, 24, 26.
Efficiency, .72, .61, .52, .44, .37, .31, .25, .19, .14, .9, .4, .0.

To obtain the highest efficiency with any fall, the dash valve should be adjusted to close at the instant the water in the drive pipe has attained its maximum velocity.

The quantity of water delivered by a hydraulic ram is approximately as follows: With from 80 to 100 feet conveyance, $\frac{1}{7}$ of the supply can be delivered at an elevation 5 times as high as the fall of the drive pipe; or $\frac{1}{15}$ of the supply can be delivered at a point about 10 times as high as the fall of the drive pipe.

The size of ram required and its connections may be found by the following table:

Quantity of Water furnished per Minute by the Spring or Brook to which the Ram is adapted.	*Caliber of Pipe.*		*Weight of Pipe (if of Lead), or if of Wrought Iron, then of ordinary weight.*		
	Drive.	*Discharge*	*Drive Pipe for any head or fall not exceeding 10 feet.*	*Discharge Pipe for not over 50 feet rise.*	*Discharge Pipe for over 50 and not exceeding 100 feet in height.*
3 qts. to 2 gals.	$\frac{3}{4}$ inch.	$\frac{3}{4}$ inch.	6 lbs. per yard.	8 lbs. per rod.	14 lbs. per rod.
1½ gals. to 4 gals.	1 "	$\frac{3}{4}$ "	8 " "	11 " "	16 " "
3 " 7 "	1¼ "	1 "	10 " "	11 " "	16 " "
6 " 14 "	2 "	1 "	23 " "	20 " "	28 " "
12 " 25 "	2½ "	1 "	40 " "	6 " per yard.	8 " per yard.
20 " 40 "	2½ or 2¾ "	1¼ "	40 to 45 "	9 " "	11 " "
25 " 75 "	4 inch.	2 "	16 lbs. per ft. C. I.	20 " "	23 " "

The length of the drive pipe should be not less than 25 feet and not more than 50 feet.

If the ram is to be placed under a greater head or fall than named in the above table, it will of course be necessary to increase the weight and strength of the drive pipe; also, if the water is to be forced to any greater height than above mentioned, the discharge pipe should be proportionately increased in weight and strength. Where the water is to be forced to any great distance (say more

than 400 yards), it is preferable to use a discharge pipe of a *larger caliber* than named in the preceding table.

With a given supply of water under a great fall, the ram is not required to be of as large a size as for the same quantity of water under a less fall.

If the stream is a large one, and a greater supply of water is required than any machine in preceding table will give, then increase the number of the machines, in preference to having one machine of a larger capacity than table gives. Several rams may be set so as to deliver into one discharge pipe, each ram having a separate drive pipe applied from spring to ram.

Size of Water Pipes in a Building.

The size of water supply pipes in a building may be found by reference to the following table:

Supply Branches.	*Low Pressure.*	*High Pressure.*
To Bath cocks	¾ to 1″	½ to ¾″
" Basin cocks	½″	½ to ½″
" W. C. flush tank	½″	¾″
" W. C. flush valve	1 to 1½″	¾ to 1″
" W. C. flush pipes	1½ to 1½″	
" Sitz or foot baths	½ to ¾″	1″
" Kitchen sinks	½ to ¾″	½ to ½″
" Pantry sinks	½″	½ to ¾″
" Slop sinks	½ to ¾″	½ to ¾″
" Urinals	½ to ¾″	½ to ¾″

Kitchen Boilers and their Connections.

Iron Boilers or Hot Water Storage Tanks should be galvanized both outside and inside, particularly inside. These boilers are commonly made of mild steel, and are not so durable as the old time wrought iron boilers. Many of the poorer grades become pitted very rapidly, and are not to be recommended on first-class work. The longitudinal seams are either *single* or *double* riveted; when double rivetted, the rivets should be *staggered*.

Ordinary boilers are tested to 150 lb. water pressure. Extra heavy boilers are tested to 250 lb. water pressure. The latter *should always* be used when the street pressure is more than 40

lb. by the gauge, or when a water hammer may at any time come upon the plumbing system.

STANDARD SIZES OF GALVANIZED BOILERS.

Capacity in U. S. Gallons.	*Length in Feet.*	*Diameter in Inches.*
18	3	12
21	3½	12
24	4	12
24	3	14
27	4½	12
28	3½	14
30	5	12
32	4	14
35	5	13
36	6	12
36	4½	14
40	5	14
42	4	16
47	4½	16
48	6	14
52	5	16
53	4	18
63	6	16
66	5	18
79	6	18
82	5	20
98	6	20
100	5	22
120	6	22
120	5	24
144	6	24
168	7	24
182	8	24

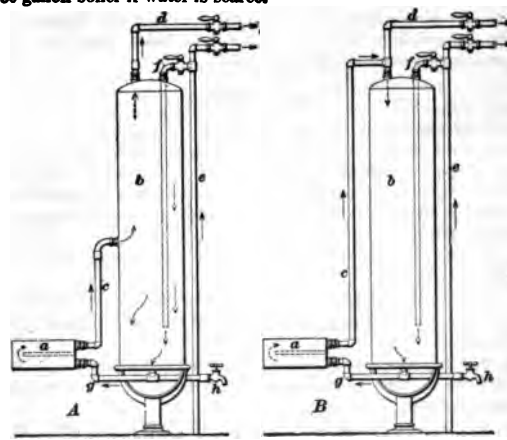
Copper range boilers are in every case to be preferred, if properly coated inside with block tin. They are classed as *light*, *heavy* and *extra-heavy*, the latter being tested to 150 lb. water pressure.

The best forms of copper boilers are those which are reinforced inside by *stiffeners* or *braces* so that they will not collapse when a partial vacuum is formed within them.

The size of boiler which should be employed for any particular work depends chiefly upon existing conditions, such, for example, as the water supply and the nature of the building. A safe

rule is to allow a 35 or 40 gallon boiler for a building having one bathroom and to add 30 gallons additional capacity for every extra bathroom. The true size of a boiler can only be computed by first ascertaining the maximum quantity of hot water which will be drawn off in a given time, and the nature of the heating agents which warm the water.

The size of the water-back is another item which cannot be computed, because it is based on the size of the boiler or the quantity of water to be heated in a given time. In practice we find that about 100 square inches of water-back heating surface in actual contact with the fire is sufficient to give good results with a 40-gallon boiler if water is plentiful, and with a 50-gallon boiler if water is scarce.



Boiler Connections.

Kitchen boilers are connected up in different ways by different people, but the most common and most reliable method is shown at *A* in the figure. The only objection to this arrangement is that it requires a long time for the water in the boiler to

become hot, because the hot water in the water-back *a* enters the side of the boiler *b* through the flow pipe *c* and mixes with the cold water when rising to the top, from which the hot water is drawn to the fixtures, by the hot water distributing pipe *d*. To overcome this trouble *c* is often connected to *d* over the boiler as shown in *B*.

The arrows show the direction in which the water flows. The cold supply *e* has a branch *f* taken off to supply the boiler through an "inner tube" within the boiler, shown by dotted lines. The return pipe *g* furnishes a supply to the water-back and the sediment cock *h* is used to empty the boiler when desired.

The chief objection to the method shown at *B* is that circulation between the range and the boiler will cease when the service pipe is shut off because the hot water will siphon down to the vent hole in the top of the inner tube. The water-back will then blow steam through *c* and *d*.

GAS FITTING.

The pipes used for the distribution of gas in buildings are standard plain wrought iron or steel pipe (see table, page 6).

If the location of the pipes is not shown by the architect then the gas fitter must use his own judgment in determining their position. He should be governed by the following considerations:

1. The pipes should run to the fixtures in the most *direct* manner practicable.

2. The pipes must be *graded* to secure proper drainage without excessive cutting of floor beams, or otherwise damaging the building.

3. Pipes which run crossways of floor beams should be laid not more than one foot away from the wall, so as to avoid serious injury to the floor.

4. Fixtures should be supplied by risers rather than drop pipes, as far as practicable.

5. All pipes should be located where they can be got at for repairs, with the least possible damage to the floors or walls.

The fittings should be malleable iron galvanized, beaded fittings being preferable to plain ones. Plain black iron fittings

are not worth shop room, and should never be used on work of any consequence.

Pressure of Gas.

If the specific gravity of a gas is less than that of air at the same temperature, then the pressure will always be greatest at the top of the pipe or chamber which contains the gas. If the gas is heavier than air, then the greatest pressure will be at the bottom of the chamber which contains it.

The upward pressure of gas having a less density than air, is caused by the deficiency in its weight, and its consequent inability to balance the pressure of the atmosphere.

For illustration, let us consider a column of gas 1 foot square and 100 feet high, having a density of .5, or one-half that of air; the temperature being the same as that of the atmosphere, say 60°. Now, air at 60° weighs 0.764 pound per cubic foot, and as the column contains 100 cubic feet, it will weigh $.0764 \times 100 = 7.64$ pounds. The gas having a density of .5 will weigh only half as much, or 3.82 pounds, and is, therefore, unable to balance an equal volume of air. Consequently, it is pressed upward with a force of $7.64 - 3.82 = 3.82$ pounds against the top of the chamber which contains it. Whatever the actual pressure of the gas may be at the bottom of the column it will, in this case, be increased at the top to the extent of 3.82 pounds per square foot.

The *increase of pressure* in each 10 feet of rise in the pipes with gas of various densities, is shown in the following table :

Rise in pressure (inches of water)	0.	.0147	.0293	.044	.058	.073	.088	.102
Density of gas	1.	.9	.8	.7	.6	.5	.5	.3

EXAMPLE.—The pressure in the basement, at the meter, is 1.2 inches of water ; what will be the pressure on the sixth story, 70 feet above, the density of the gas being .4 ?

SOLUTION.—The table shows that the increase will be .088 for *each 10 feet of rise* ; therefore, $.088 \times 7 = .616$ increase. Then *pressure at sixth story* $= 1.2 + .616 = 1.816$ in.

Measuring Pressure of Gas.

The pressure of gas is measured by the common *water gauge*, which is shown in Figure. The tubes *b* and *c* are made of glass, and are filled with water up to the zero of the scale. The scale is graduated in inches and convenient fractions of an inch. The tube *c* is open to the air at the top. When pressure is admitted to *a*, the water will sink in the tube *b*, and will rise in *c*. The difference in the height of the water in the two tubes, measured in inches, is the measure of the pressure exerted in *inches of water*.

For measuring heavier pressures, mercury is used in the tubes, instead of water.

Pressures which have been measured in inches of water or mercury, may be translated into pounds per square inch or square foot, by multiplying the reading by the following figures :

One inch of water at 62° = 5.2 lb. per sq. foot.
" " " " " = .0361 " " " inch.
" " " mercury " = .4897 " " " "

Pressure per square inch or square foot may be converted into inches or feet of water, or inches of mercury, by multiplying the pressures by the following figures :

One pound per square foot = .1923 inches of water at 62°.
" " " " inch = 27.7 " " " "
" " " " inch = .2042 " " mercury "

Measuring the Flow of Gas.

The *velocity* of gas flowing through a pipe may be measured by means of *Pitot's tube*.

The volume of gas passing through a pipe in a given time is *computed by* multiplying the velocity, as found by Pitot's tube, *by the area of* the pipe.

The actual *quantity* of the gas is computed by correcting the volume for temperature and pressure, reducing it to a volume at standard temperature of 32°, and standard pressure of 1 inch of water. The correction for temperature may be made by use of the following

Rule.—Multiply the measured volume by the actual temperature plus 460, and divide the product by 492. The quotient will be the volume at 32°.

The correction for pressure may be made by aid of the following

Rule.—Multiply the volume by the pressure in inches of water plus 407, and divide the product by 408. The quotient will be the volume at 1 inch pressure.

EXAMPLE.—A certain pipe passes 1,000 cubic feet of gas per hour, under a pressure of 8 inches of water; what will the volume be when the pressure is reduced to 1 inch, the temperature remaining the same?

SOLUTION.—Applying rule:

$$1{,}000 \times (8 + 407) \div 408 = 1{,}017.1 \text{ cu. ft.}$$

In dealing with gas it is necessary to keep in mind a clear distinction between the *apparent volume* of the gas, and the *actual quantity*. The former is the volume as measured at the actual temperature and pressure, while the latter is the volume of the same gas at standard temperature and pressure.

For ordinary purposes, the volume of gas passing through a pipe is measured by an apparatus called a *gas meter*. A gas meter measures the *volume* only, and its indications are not affected by any change that may occur in the pressure of the gas. The difficulty thus encountered in correctly measuring the volume of gas actually delivered under varying pressures, is easily overcome by using a *governor* between the meter and the street main or service pipe. The governor is a species of reducing valve which will receive gas at any pressure, whether *steady or variable*, and will discharge it at a steady low pressure.

Reading Gas Meters.

The figure shows a diagram of a meter dial of the ordinary type.

To read a meter dial of this description, first write down the figure which the pointer has just passed on each dial, then add two ciphers to the right and the number so obtained will be the amount of gas in cubic feet which the meter has measured.

Thus, the pointers on the diagram indicate that 14,200 cubic feet of gas has passed through the meter. When the pointer to

the left has made one complete revolution, the process of indicating is repeated. The pointers all move from the smaller to larger figures, just like the hands of a clock.

Pressure.

The objects sought in the use of pressure regulators or governors, are, first, *economy* in the consumption of gas, and, secondly, *steadiness* of the lights and most *effective operation* of the burners.

It is of great importance that both the volume and pressure at the burners should be closely regulated. The amount of gas wasted by over pressure is much greater than is generally believed. A good new lava-tip burner consuming 5 cubic feet per hour at .5 inch pressure, will consume about .5 of a cubic foot *more for each* increase of .1 inch in the pressure. Thus an over *pressure of .1* inch will increase the gas bill about 10 per cent.

The variation in the gas pressure, even in the best regulated systems, is usually much greater than one-tenth inch, frequently ten-tenths and more.

There are two systems of gas regulation now in use, namely : *pressure regulation* and *volumetric regulation*. In the first system a governor is attached to the service pipe at the meter, and the house distributing pipes are maintained at a constant amount ; in the second system, *each burner* is supplied with a governor, and the pressure in the house pipes is not controlled, being about the same as in the mains.

The *proper place* for a pressure regulator, if one be used, is between the meter and the main. The meter should then be adjusted to suit the house pressure instead of the street pressure.

The pressure which should be given to the gas at the burners, in order to secure the best results, varies greatly in different forms of apparatus. The following are the pressures generally used :

Argand burners..... .2 inch of water.
Common Batswing burners..... .5 " "
Wellsbach incandescent burners..... .5 or more.
Wenham and Lebrun lamps ..... .5 to 1 or more.
Atmospheric burners.....1.0 or more.

CAPACITY OF GAS PIPES.

Diameter of Pipe. Inches.	*Maximum Length. Feet.*	*Capacity Per Hour.*	
		Coal Gas. Cubic Feet.	*Gasoline Gas. Cubic Feet.*
¼	6	10	
⅜	20	15	10
½	30	30	20
¾	50	100	75
1	70	175	125
1¼	100	300	200
1½	150	500	350
2	200	1,000	700
2½	300	1,500	1,100
3	450	2,250	1,500
4	600	3,750	2,500

The pressure of the gas is assumed in the foregoing table to be about two inches of water. It should be understood that the quantities stated are those which the pipes will deliver at the burners *without objectionable fall of pressure.*

Testing a System of Pipes.

As soon as the pipes are all in place and are properly secured, the system should be tested to find if it is perfectly gas-tight.

Air should be forced in the system until the gauge indicates 15 or 20 inches of mercury, or 7 to 10 pounds per square inch. The pressure should be continued in the pipes for about an hour, and if the gauge shows a falling off in pressure of more than one-fourth inch of mercury, or one-eighth of a pound per square inch, then the system cannot be passed as perfect.

The extent of the leak may be judged by the rapidity of the fall in pressure, but its location must be found by the sense of smell.

For this purpose a small quantity of ether should be introduced into the pipes. The pressure gauge should be provided with an ether cup especially for testing purposes. The odor of the ether will diffuse throughout the system of piping and will escape from the leak.

In case of large buildings, it is advisable to test the piping in sections, say one floor at a time, because it is much easier to locate leaks. After each section is tested they may be connected, and then subjected to a final test.

The pipes should not be covered until the tests are completed. The owner, architect or inspector should witness the tests.

HEATING AND VENTILATION.

VENTILATION is a process of moving foul air from any space and replacing it with fresh air. A positive displacement, however, does not take place ; the incoming fresh air chiefly dilutes the foul air to a point suitable for healthful respiration.

Pure air such as exists in the open country contains about 4 *parts of* carbonic acid per 10,000 parts of air, while badly ventilated rooms often contain as much as 80 parts of CO_2 per 10,000

of air. Hygienists, after careful study, have decided that an increase of 2 parts of $C O_2$ per 10,000 of air should be accepted as the *standard of respirable purity*, and any excess of $C O_2$ above this may be considered as vitiation.

The $C O_2$ test for vitiation of air, although not scientifically correct, is the best which practical men can make, because the amount of $C O_2$ present is a fair indication of the true condition of the air.

Sources of Vitiation.

The average adult exhales about 0.6 cu. ft. of $C O_2$ per hour.

One cu. ft. of gas, while burning, gives out about .8 cu. ft. of $C O_2$.

An ordinary lamp gives out about 1.0 cu. ft. of $C O_2$ per hour.

One candle " " " .3 " " "

To compute the quantity of air required to maintain any desired degree of purity by diluting foul with fresh air,

Let Q = quantity of fresh air to be supplied per hour.

c = cubic feet of $C O_2$ in each 10,000 cubic feet of the entering air.

C = cubic feet of $C O_2$ which each 10,000 cubic feet of the air in the room may contain for proper health conditions.

n = number of persons in the room.

Then $\frac{Q \times c}{10,000} + .6\,n$ = cu. ft. of $C O_2$ given off to the room in one hour, which, if divided by Q and multiplied by 10,000, will give the proportion of $C O_2$ in 10,000 parts of the air in the room. This should be equal to C, the standard of purity desired.

Therefore, $C = \frac{10,000\left[\frac{Q \times c}{10,000} + .6\,n\right]}{Q}$, or $Q = \frac{6,000\,n}{C - c}$.

Representing c by 4, and C by 6, $Q = \frac{6,000}{6 - 4}n = 3,000\,n$; that is to say, it requires 3,000 cu. ft. of fresh air per person to

maintain a respirable standard of 6 parts CO_2 in 10,000 parts of air.

EXAMPLE.—A room contains 40 people; how much fresh air is required per hour to maintain a respirable purity of 8 parts CO_2 per 10,000 parts of air?

$$\text{SOLUTION.—}Q = \frac{6{,}000\ n}{C - c} = \frac{6{,}000 \times 40}{8 - 4} = 60{,}000 \text{ cu. ft.}$$

If any lights deliver their products of combustion into the room, the amount of CO_2 given off by them should be converted into its equivalent in men, thus: One ordinary gas light equals in vitiating effect about 5½ men, an ordinary lamp 1⅔ men and an ordinary candle about ⅙ man.

It is considered good practice to allow 2,000 cu. ft. of fresh air per hour for each inmate of a room or auditorium. Hospitals and such places where the vitiation is due to exhalations from diseased or sick people should be provided with at least twice this amount.

Forced Ventilation.

There are two classes of forced ventilation: (1) The *plenum system*, that in which the pressure of the air in the building is slightly greater than that of the outer atmosphere, in other words, that system by which air is blown through the building by means of a fan or other blower placed at the inlet. (2) The *vacuum system*, or that method of removing foul air from a building and causing a consequent inrush of fresh air from the outside, by means of an exhaust fan placed at the outlet to the vent flue or stack. In the latter system the pressure of the air in the building is slightly lower than that of the outer atmosphere. The plenum system is the more wholesome method, and is to be preferred in ventilating buildings.

The best proportions for centrifugal fans for this work are shown in the figure. The numerals denote the sizes of the different parts compared with the diameter of the fan wheel, which is taken as a unit.

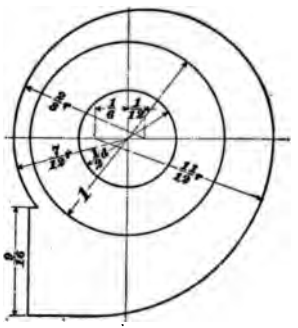
The housing, or outer casing, is one-half the diameter of the wheel, and three-eighths for narrow fans.

Taking centers and radii as shown by broken lines, a very close approximation to the actual curves can be obtained.

The capacity of a fan, that is to say, the amount of air which a fan will deliver, depends considerably upon the construction of the fan, and upon the resistance to the flow of the air.

Fresh-air inlets, if near the floor, should be so arranged that the velocity of inlet will not exceed 2 feet per second; a higher velocity is considered a perceptible draft. Higher velocities, however, are permissible if the inlets are located more than 8 feet above the floor.

Vent-flue inlets can safely be made to operate with a velocity of 6 or 8 feet per second, without being disagreeably perceptible.



In natural ventilation systems, the drafts in the flues or ducts are caused by the difference in density between the air in the ducts and the outer atmosphere. The higher the temperature of the ducts, the more rapid will the draft become.

Velocity of Flow of Air in Hot Air Ducts.

Let v = the theoretical velocity in feet per second of the air flowing through a duct.
h = the vertical height in feet of the duct.
t = absolute temperature of external air.
t_1 = absolute temperature of the heated air.

The *theoretical velocity* can be computed approximately by the formula $v = 8.02\sqrt{\frac{h\,(t_1-t)}{t}}$.

The *actual velocity* will, however, be considerably less. It will vary with the resistance to the flow of the air, due to skin friction, bends, etc. The resistance will also vary with the cross-section of the duct, and with the forms of connections used.

If the loss by friction, etc., is estimated at 50%, which is a fair average in good work, the formula $v = 4\sqrt{\frac{h\,(t_1 - t)}{t}}$ will give a close approximation.

Knowing the actual velocity in a duct, the quantity of air discharged in cubic feet per minute can be found by multiplying the sectional area of the duct, in square feet, by the velocity of the air in feet per minute.

The following table shows a safe capacity for some leading forms of blowers and exhaust fans, operating against a pressure of 1 oz. per sq. in. or 1¾" water, nearly.

Diameter of Wheel, Feet.	*Revolutions per Minute.*	*Horsepower Required.*	*Capacity in Cubic Feet per Minute.*
4	350	6.0	10,600
5	325	9.4	17,000
6	275	13.5	29,600
7	230	18.4	42,700
8	200	24.0	46,000
9	175	29.0	56,800
10	160	35.5	70,300

If the resistance to the fan is greater than that named in the table, the same volumes delivered may be obtained by increasing the revolutions of the wheel and by increasing the horsepower correspondingly. Or, if the resistance is less, the same capacities can be obtained by reducing the number of revolutions per minute, which means a corresponding decrease of power required.

HEATING BUILDINGS.

The apparatus required for heating buildings with steam or hot water, is necessarily composed of a heater, a series of pipes which convey the steam or hot water from the heater, and a number of radiators to heat the air in the building.

Steam Heating.

When a building is heated with steam having a pressure less than 10 lb. by the gauge it is said to be heated by a *low pressure system.*

If heated by steam at a higher pressure, the system is known as a *high pressure system.*

When the water of condensation in a heating system flows back to the boiler by the force of gravity alone, the apparatus is known as a *gravity circulating system.*

When the boiler is run at a high pressure and the heating system at a low pressure the water of condensation must be returned to the boiler by the aid of mechanical contrivances such as pumps, steam return traps, or injectors.

Low pressure gravity circulating apparatus may be erected in different systems, but those considered best practice are: the *two pipe system with wet returns; the two pipe system with dry returns; the one pipe system, in which all mains, branches and risers are relieved into a return main below the water-line; and the one pipe circuit system, in which all pipes are run above the water-line.* The choice of any system should depend upon special conditions and requirements which will govern the work under consideration.

Radiation.

To find amount of direct radiation required to heat a given room, basing calculations upon the cubic contents of the room, allow 1 sq. ft. direct radiating surface to the number of cubic feet shown in the following table:

	Cu. Ft.
Bath room, living room, with 2 or 3 exposures and large amount of glass	40
Living room, one or two exposures with large amount of glass	50
Sleeping rooms	55 to 70
Halls	50 to 70
School rooms	60 to 80
Churches and auditoriums having large cubic contents and high ceilings	65 to 100
Lofts, workshops and factories	75 to 150

The above ratios will give reasonably good results on ordinary

work, if the engineer uses proper judgment in allowing for exposures, leakages through building, etc.

PROPORTION OF RADIATING SURFACE TO GLASS SURFACE.

BALDWIN'S RULE.—"*Divide the difference in temperature between that at which the room is to be kept and the coldest outside atmosphere, by the difference between the temperature of the steam pipes and that at which you wish to keep the room, and the quotient will be the square feet or fraction thereof of plate or pipe surface to each square foot of glass, or its equivalent in wall surface.*"

The quantity of heating surface found by this simple rule only compensates for the amount of heat lost by transmission through the windows, walls and other cooling surfaces, which it is calculated against; it does not provide for cold air entering the room through loosely fitting doors, windows, etc., and an ample allowance must be made for this. Some buildings are so poorly constructed that 50 per cent. or more must be added to the amount of heating surface obtained by the above rule in order to counteract the cooling effect of these air leakages. A common practice is to add 25 per cent. for buildings of ordinary good construction. Ample allowance should be made for rooms exposed to cold winds.

Let S = amount of radiating surface required to counteract the cooling effect of the glass and its equivalent in *exposed* wall surface in square feet.

t = difference in degrees F. between the desired temperature of the room and that of the external air.

t_1 = difference in degrees F. between the temperature of the heating surface and that of the air in the room.

s = number of square feet of glass and its equivalent in exposed wall surface.

Then, applying rule, $S = \frac{t}{t_1} s$.

It is common practice to estimate about 10 square feet of wall surface to be equivalent in cooling power to one square foot of glass.

Radiators.

SIZES OF TAPPINGS FOR PRIME SURFACE DIRECT RADIATORS.

One Pipe Work.

Less than 25 sq. ft.	1	inch
From 25 to 50 sq. ft.	1¼	"
" 50 " 90 "	1½	"
" 90 " 160 "	2	"

Radiators.—*Continued.*

Two Pipe Work.	*Steam.*	*Return.*
Less than 30 sq. ft.	¾ inch	¾ inch
From 30 to 50 sq. ft.	1 "	¾ "
" 50 " 100 "	1¼ "	1 "
" 100 " 160 "	1½ "	1¼ "

Indirect Radiation.	*Steam.*	*Return.*
Less than 30 sq. ft.	1 inch	¾ inch
From 30 to 50 sq. ft.	1¼ "	1 "
" 50 " 100 "	1½ "	1¼ "
" 100 " 160 "	2 "	1½ "

Sizes of Steam Pipes.

Radiation.	*One Pipe Work.*	*Two Pipe Work.*	
		¾ in. steam	¾ in. return
40 to 50 sq. ft.	1 inch	1 "	¾ "
100 " 125 "	1¼ "	1¼ "	1 "
125 " 250 "	1½ "	1½ "	1¼ "
250 " 400 "	2 "	2 "	1½ "
400 " 600 "	2½ "	2½ "	2 "
600 " 900 "	3 "	3 "	2½ "
900 " 1,200 "	3½ "	3½ "	3 "
1,200 " 1,600 "	4 "	4 "	3½ "
1,600 " 2,000 "	4½ "	4½ "	4 "
2,000 " 2,500 "	5 "	5 "	4½ "
2,500 " 3,600 "	6 "	6 "	5 "
3,600 " 5,000 "	7 "	7 "	6 "
5,000 " 6,500 "	8 "	8 "	6 "
6,500 " 8,000 "	9 "	9 "	6 "
8,000 " 10,000 "	10 "		

If the mains are high above the boiler, and if they have short straight runs, small pipes may be used. If they are unusually low or unusually long or crooked, they should be large. The above table will give good results in ordinary systems.

Size of Steam Boilers.

No hard and fast rules can be employed for determining the heating capacity of boilers or for ascertaining the sizes of boilers best adapted for different jobs.

The following table is deduced from practical tests and observations by Prof. Carpenter, of Cornell University, and is considered to be within the limits of safety.

Two cases are considered in the table; (A) when the rate of coal consumption is 10 lb. and (B) when it is 8 lb. per square ft. *of grate surface* per hour. The latter is preferable for hot-*water heating.*

Radiation by Feet	250	500	750	1000	1500	2000	3000	4000	5000	7500	10000
Nominal horsepower	2.5	5	7.5	10	15	20	30	40	50	75	100
Ratio radiating to heating surface	1.5	1.5	1.5	1.5	1	1	1	1	1	1	1
Evaporation per lb. of coal	5.5	5.5	6	6	6	6	6	6	6	6	6
Pounds of steam per sq. ft. grate (A)	55	55	60	65	70	75	80	85	90	105	120
" " " " " (B)	11	10	10	13	14	15	16	17	18	21	24
Ratio radiating to grate surface (A)	165	165	180	195	210	225	240	255	270	300	330
" " " " " (B)	132	132	144	156	168	180	192	204	216	240	264
" heating " " (A)	30 h	33 h	36 h	39 h	42 h	45	48 h	51 h	54 h	60 h	66 h
" " " " (B)	24 h	26 h	28 h	31 h	34 h	36	38 h	41 h	43 h	48 h	53 h
Boiler heating surface, sq. ft.	55	100	150	195	300	450	600	780	900	1350	1800
Grate surface, sq. ft. (A)	1.5	3	4.15	5	7.15	10	15	19	25	35	45
" " " (B)	1.88	3.88	5.4	6.25	8.93	11.3	15 h	19 h	23	32	41 h
Diameter safety valve, inches	1 h	2	2	2	2	2	3	3	4	4	4
" smoke pipe "	7	10	11	12	15	17	20	24	26	30	34

Hot Water Heaters.

Ratio radiating to heating surface	6	6	6.1	6.4	6	6.3	10	10	10	10 h	10 h
" " " " grate " (A)	247	250	270	290	315	337	360	383	405	487	540
" " " " " (B)	188	187	210	220	240	270	280	300	324	378	420
" heating " " (A)	30 h	33	36	39	42	45	36	38	40	48	54
" " " " (B)	24 h	25	28	30	32	36	28	30	32	38	42
Heating surface, sq. ft.	30	55	91	118	180	245	360	465	550	825	1100
Grate surface, sq. ft. (A)	1	1.10	2.25	3.75	4.75	5.9	8	10	13	17	22
" " " (B)	1.25	2.58	3.8	4.25	5.9	7.4	10	13	16	21	27
Diameter smoke flue, inches	7	10	11	12	15	17	20	24	26	30	34

* Water tube boiler.

Exhaust Heating.

Exhaust steam from non-condensing engines usually contains from 20 to 25 per cent. of water and oil, the latter being employed to lubricate the engine cylinders. Before exhaust steam is allowed to enter a heating system, the water and oil should be separated from it.

The effect of turning exhaust steam into a heating system is to form a back-pressure on the engine. This must be avoided as much as possible by using large steam-distributing pipes.

A direct connection to the steam boilers through a pressure-reducing valve must be employed, to automatically furnish steam to the heating system when the exhaust fails. A relief valve, also, should be placed upon the system, so that surplus exhaust steam may escape to the atmosphere.

To proportion an exhaust-heating system, it is necessary to know about how many square feet of radiating surface we should employ to properly condense the exhaust steam from the non-condensing engines. To do this we must first know the weight of steam which would be discharged from the engine.

Class of Non-Condensing Engine.	*Water used per hour for indicated Horsepower.*
Compound Automatic	25 lbs.
Simple Corliss	30 "
Simple Automatic	35 "
Simple Throttling	40 "

From this must be deducted about 10 per cent. for condensation in the cylinders, etc., in order to obtain the real available weight of steam for heating purposes.

APPROXIMATE RATIO BETWEEN CUBIC CONTENTS AND RADIATOR SURFACE FOR EXHAUST HEATING.

Class of Building.	*Direct Radiation.*	*Indirect Radiation.*	*Blower System.*
	sq.ft. cu.ft.	*sq.ft. cu.ft.*	*sq.ft. cu.ft.*
Dwellings	1 to 50	1 to 40	1 to 300
Offices	1 " 70	1 " 60	1 " 895
Stores and shops	1 " 100	1 " 80	1 " 600
Churches, etc	1 " 200	1 " 150	1 " 900

The figures on preceding page simply form a reasonable average, and allowance must be made for exposure, etc.

Each square foot of direct radiating surface gives off to the air around it about 1½ Thermal Units per hour per degree of difference between the temperature of the steam and that of the surrounding air. This is eqvivalent to about ¼ of a pound of steam per hour, or, in other words, about 4 to 4¼ square feet of surface to each pound of steam to be condensed.

HOT WATER HEATING.

Circulation.

Circulation of water in a hot water-heating system is a movement of hot water from the boiler towards the radiators where it parts with some heat, and a consequent movement of colder water from the radiators to the boiler to become reheated. Without circulation, we cannot convey heat from the boiler to the radiators.

The velocity of circulation depends chiefly upon: (a) the difference between the mean density of the water in the ascending current and the mean density of the water in the returning current; (b) the vertical height of circuit above the boiler; (c) the resistance to the flow due to friction, change in direction, vena-contracta, etc.

The *theoretical velocity* of circulation can easily be computed by formulas, but we refrain from giving any, because, first, the theoretical velocity is only an imaginary thing and of little benefit to the practical man; and, second, because the *actual velocity* bears no definite ratio to the theoretical.

The sizes of pipe should be governed by the amount of radiating surface they have to supply, the height of the radiators above the boiler, and the number of changes in the direction of the several currents.

It is considered safe practice to allow from 50 to 100 square feet of direct radiation for every square inch of cross-section in the pipe.

If the pipes are short, straight and high, 1 to 100 would be allowed; if long, crooked or low, 1 to 50 or more, according to the condition, would be employed.

The sectional area of the mains should approximate the sum of the area of its branches, but in many cases, particularly where there is no indirect work, the sizes may be a little less.

SIZES OF MAINS AND BRANCHES.

Sizes of Main.		*Sizes of Branches.*
1"	will supply two	¾".
1¼"	"	two 1", or one 1" and two ¾".
1½"	"	two 1¼", or one 1¼" and two 1".
2"	"	two 1½", or one 1½" and two 1¼".
2½"	"	two 1½" and one 1¼", or one 2" and one 1¼".
3"	"	one 2½" and one 2", or two 2" and one 1½".
3½"	"	two 2½", or one 3" and one 2", or three 2".
4"	"	one 3½" and one 2½", or two 3", or four 2".
4½"	"	one 3½" and one 3", or one 4" and one 2½".
5"	"	one 4" and one 3", or one 4½" and one 2½".
6"	"	two 4" and one 3", or four 3", or ten 2".
7"	"	one 6" and one 4", or three 4" and one 2".
8"	"	two 6" and one 5", or five 4" and two 2".

All flow and return pipes should be of the same size.

All pipes must rise from the boiler to the radiators with a pitch of at least 1" in 10 ft.

To determine the amount of radiating surface necessary to easily warm a building in all kinds of weather, and to proportion it so that each room will have the required temperature at the same time, is a very important item and must be carefully considered.

No formula, rule or table, can be given which will be applicable to all cases, but for ordinary buildings having the average wall and glass exposures, the table on following page is found in practice to give good results, when used with good judgment.

Ratio of Hot Water Radiating Surface to Cubical Contents of Rooms.

Dwellings.	*One sq. ft. of Direct Radiating Surface will Heat.*
Living rooms, one side exposed	30 cu. ft.
" " two sides exposed	28 "
" " three sides exposed	25 "
Sleeping "	From 30 to 40 cu. ft.
Hall "	" 20 " 30 "
Bath "	
Public Buildings.	
School rooms	From 30 to 50 cu. ft.
Offices	" 50 " 70 "
Factories	
Stores	
Auditoriums	" 80 " 100 "
Churches	

The above ratios are for an average temperature of 160° Fah. in the radiators.

For direct-indirect radiation allow not less than 25 per cent. extra.

For indirect radiation allow at least 50 per cent. additional surface.

Due allowance must be made for leakages through loose doors, windows, etc.

DIRECT HOT WATER RADIATOR TAPPINGS.

Heating Surface.	*Size of Tapping.*	*Location of Radiator.*
0 to 15 sq. ft. tapped	¾ inch	First floor.
15 " 50 " "	1 "	
50 " 100 " "	1¼ "	
100 and over " "	1½ "	
0 to 20 " "	¾ "	Second floor.
20 " 70 " "	1 "	
70 " 125 " "	1¼ "	
125 and over " "	1½ "	

INDIRECT RADIATOR TAPPINGS.

0	to	40	sq. ft. tapped	1	inch.
40	"	90	" "	1¼	"
90	"	150	" "	1½	"
150	and	over	" "	2	"

When extended surface indirect radiators are used, allow 30 per cent. more than the manufacturer's ratings.

The expansion tank capacity should be at least $\frac{1}{10}$ of that of the entire apparatus, if it is an open tank. We do not advise the use of closed expansion tanks.

COMPASS SURVEYING.

The *bearing* of a line is the angle which it makes with the direction of the magnetic needle. The length of a line, together with its bearing, is termed the *course*. To take the bearings of a line, set the compass directly over a point in it, at one extremity, if possible. This may be done by means of a plumb bob suspended from the compass.

Bring the compass to a perfectly level position. Let a flagman hold a rod carefully plumbed at another point of the line, preferably the other extremity, if he can be distinctly seen. Direct the sights upon this rod and as near the bottom of it as possible. Always keep the same end of the compass ahead; the north end is preferable, as it is readily distinguished by some conspicuous mark, usually a *fleur de lis*, and always read the same end of the needle, that is, the north end of the needle if the north point of the compass is ahead, and *vice versa*. Before reading the angle, see that the eye is in the direct line of the needle so as to avoid the error which would otherwise result from *parallax*, or apparent change of the position of the needle, due to looking at it obliquely.

The angle is read and recorded by noting, *first*, whether the N. or S. point of the compass is nearest the end of the needle being read; *second*, the number of degrees to which it points, and *third*, the letter E. or W. nearest the end of the needle being read.

Let *A B* in Fig. 1, be the direction of the magnetic needle, *B* being at the north end. Let the sights of the compass be

directed along the line *C D* The north point of the compass will be seen to be nearest the north end of the needle which is to be read. The needle, which has remained stationary while the sights were being turned to *C D*, now points to 45° between the *N* and *E* points, and the angle is read north forty-five degrees east (N. 45° E.).

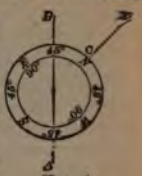


FIG. 1.

A sure test of the accuracy of a bearing is to set up the compass at the other end of the line, *i. e.*, the end first sighted to, and sight to a rod set up at the starting point. This process is called *back-sighting*. If the second bearing is the same as the first, the reading is correct. If it is not the same, it shows that there is some disturbing influence at either one or the other end of the line. To determine which of these two bearings is the true one, the compass must be set up at one or more intermediate points, when two or more similar bearings will prove the true one.

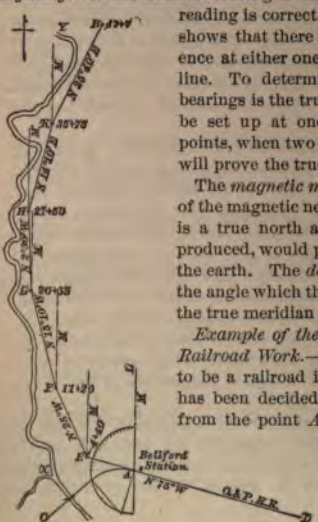


FIG. 2.

The *magnetic meridian* is the direction of the magnetic needle. The *true meridian* is a true north and south line, which, if produced, would pass through the poles of the earth. The *declination of the needle* is the angle which the magnetic meridian and the true meridian make with each other.

Example of the Use of the Compass in Railroad Work.—Suppose *C A D* in Fig. 2 to be a railroad in operation, and that it has been decided to run a compass line from the point *A* along the valley of the stream *X Y* to the point *B*. The bearing of the tangent *A D* cannot be determined by setting up the compass at *A* on account of the attraction

of the rails. The *direction* of this tangent, however, can be obtained by setting up at *A* and sighting to a flag held at *D*. The point *A*, which is the starting point of the line to be run, is marked *O*. Producing the line *A D* 440 feet, the point *E* is reached, which has been previously decided upon as a proper place for changing the direction of the line. The compass being set up at *E*, the bearing of the line *A E*, which is the line *A D* produced, is found by sighting to *A*, or, what is preferable, to the point *D*, if that point can be seen. The number of Station E, *viz.*, 4 + 40, and the bearing of *AE* is then recorded by the compassman. By this time the chief of party has located the point *F*, and the flag is in place for sighting. The axemen, if there is work for them to do, are put in line by the head chainman, clearing only so much as would interfere with rapid chaining. The bearing of the line *E F* being recorded, the compass is moved quickly to *F*, replacing the target left by the flagman, leveled up and directed toward the point *G*, which is already located. The chainmen reaching *F*, its number 11 + 20 is recorded by the compassman and the instrument sighted to *G* and the work continued as before.

Form for Keeping Notes.

A plain and convenient form for compass notes is the following, which is a record of the survey platted in Fig. 2: The first column contains the station numbers, the notation running from the bottom to the top of the page. By such an arrangement, the lengths of the courses are found by subtracting the number of the station of one compass point from the number of the station of the next succeeding compass point. Before commencing the plat, the subtractions are made and the lengths of the courses written in red ink between the station numbers.

The second column contains the bearings of the lines. The bearing recorded opposite to a station is the bearing at the course between the given station and the one next above. Thus, the bearing recorded opposite Sta. 0 is 75° 00′ W., and is the bearing of the line extending from Sta. 0 to Sta. 4 + 40 next above. The length of the course is the difference between 0 and 4 + 40 equal to 440 ft. The bearing recorded opposite to 4 + 40

is N. 25° 00′ W. It is the bearing of the line extending from Sta. 4 + 40 to Sta. 11 + 20 next above. Its length is found by subtracting 4 + 40 from 11 + 20 equal to 680 ft. and so on.

In the third column, under the head of remarks, are recorded notes of reference, topography and any information which may aid in platting or subsequent location.

Station.	*Bearing.*	*Remarks.*
47 + 75		End of line.
35 + 75	N. 25° 40′ E.	
27 + 50	N. 14° 10′ E.	
20 + 35	N. 2° 30′ W.	Woodland.
11 + 20	N. 15° 10′ W.	
4 + 40	N. 25° 00′ W.	
0	N. 75° 00′ W.	Sta. 0 is at P. C. of 14° curve to left at Bellford Sta. O. & P. R. R.

TRANSIT SURVEYING.

The Vernier.

A *vernier* is a contrivance for measuring smaller portions of space than those into which a line is actually divided. The divided circle of the transit is graduated to half degrees or 30′. The graduations on the verniers run in both directions from its zero mark, making two distinct verniers, one for reading angles turned to the right and the other for reading those turned to the left.

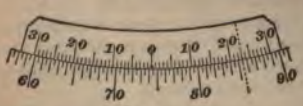


FIG. 1.

In reading the vernier, the observer should first note in which direction the graduations of the divided circle run. In Fig. 1, the graduations increase from left to right and extend from 57° to 91°. Next he should note the point where the zero mark of the vernier comes on the divided circle. In Fig. 1, the zero mark comes between 74° and

74½°. Now, as the circle graduations read from left to right, we read the right-hand vernier and find that the 23d graduation on the vernier coincides with a graduation on the divided circle and the vernier reads 23′, which we add to 74° making a reading of 74° 23′, an angle to the *left*. In Fig. 2 the graduations on the circle increase from right to left, and we accordingly read the left-hand vernier. The zero mark of the vernier comes between 67½° and 68°. Reading the vernier, we find that the 13th graduation on the vernier coincides with a graduation on the circle and the vernier reads 13′. Accordingly, we add to 67½°, the reading = 13′ making a total reading of 67° 43′, an angle to the *right*.

Setting Up the Instrument.—In setting up a transit, three preliminary conditions should be met as nearly as possible, *viz.*:

1.—The tripod feet should be firmly planted.

2.—The plate on which the leveling screws rest should be level.

3.—The plumb bob should be directly over the given point.

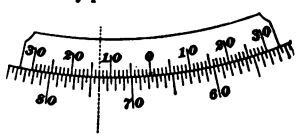


FIG. 2.

When these three conditions are met, the completion of the operation is quickly performed with the leveling screws.

How to Prolong a Straight Line.—Let *A B*, in Fig. 3, be a straight line which it is required to prolong or "produce".

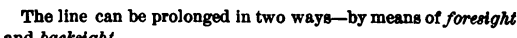


FIG. 3.

The line can be prolonged in two ways—by means of *foresight* and *backsight*.

1.—By foresight, set up the transit at *A* and sight to *B*; measure 400 feet from *B* in the opposite direction from *A*. Then, by means of signals, move the flag to the right or left until the vertical cross-hair shall exactly divide the flag held at *C*. Then, the line *B C* will be the prolongation of the line *A B*.

2.—By backsight, set the transit at *B* and sight to *A*. Reverse the telescope, and having measured 400 feet from *B* in the oppo-

site direction from *A*, set the flag at *C*, then will the line *B C* be the line *A B* produced.

Horizontal Angles and Their Measurements. — A horizontal angle is one, the boundary lines of which lie in the same horizontal plane. Let *A*, *B* and *C*, in Fig. 4, be three points, and let it be required to find the horizontal angle formed by the lines *A B* and *A C* joining these points. Set up the instrument precisely over the point *A*, and carefully level it. Set the vernier at zero, and place flags at *B* and *C*. Sight to the flag at *B* and set the lower clamp. Then, by means of the lower tangent screw, cause the vertical cross-hair to exactly bisect the flag at *B*. Loosen the upper clamp. With a hand on either standard, turn the telescope in the same direction as that of the hands of a watch until the flag at *C* is covered or nearly covered by the vertical cross-hair. Clamp the upper plate and with the upper tangent screw bring the line of sight exactly on the flag at *C*. The arc of the graduated circle traversed by the zero point of the vernier will be the measure of the angle *B A C*, as 143° 30′. The points *A*, *B* and *C* are not necessarily in the same horizontal plane, but the level plate of the instrument projects them into the horizontal plane in which it revolves.

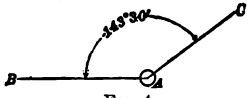


FIG. 4.

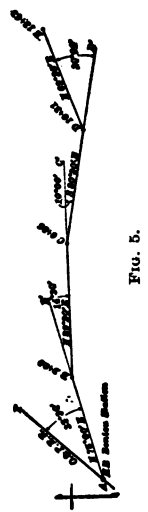


FIG. 5.

A Deflected Line. — A deflected line or "angle line" is a consecutive series of lines and angles. The direction of each line is referred to the line immediately preceding it, the latter being, in imagination, produced and the angle measured between it and the next line actually run. The angles are recorded R↱ or L↰, according as they are turned *to the right* or left of the prolongation of the *immediately* preceding line. An example of a deflected line is

shown in Fig. 5. Starting from the head block of switch at Benton Station, O. & P. R. R.

Set up the transit at *A* with vernier at zero. Sight to a flag held at *F* on the center line of the track, O. & P. R. R. Loosen the vernier clamp, the point *B* being determined, and turn the telescope until the point *B* is distinctly seen; clamp the vernier, and accurately sight to flag held at *B*; the angle reads 32° 30′ and is recorded R$_T$ 32° 30′, with a sketch showing the connection. The bearing of the line *A B* cannot be taken at *A* on account of the attraction of the rails. The point *A* is in the head block of switch (which is designated by the abbreviation *H. B.*) at Benton Station, O. & P. R. R. The instrument is now moved to *B*, the vernier set at zero and backsighted to *A*; the bearing of *A B*, *viz.*, N. 75° 00′ E., is taken, and the number of station *B*, *viz.*, 2+90, together with the bearing of *A B* recorded. The telescope is then reversed, pointing in the direction *B B′*. The point *C* being determined, the upper clamp is loosened and the telescope turned to the right and sighted to *C*. The reading is found to be 14° 30′ and recorded R$_T$ 14° 30′. It measures the angle *B′ B C*. The bearing N. 89° 20′ E. is then recorded. The instrument is next set up at *C*, the vernier set at zero, backsighted to *B*, and then reversed; the deflection to *D*; *viz.*, R$_T$ 10° 00′ read and recorded, together with the number of the station at *C*, *viz.*, 6+85. This deflection measures the angle *C′ C D* and gives the direction of the line *C D*. A good form of notes for such a survey is the following.

Station.	*Deflection.*	*Mag. Bearing.*	*Ded. Bearing.*	*Remarks.*
13+63				*End of Line.*
10+31	*L$_T$ 30°00′*	*N. 69°25′ E.*	*N. 69°30′ E.*	*Crossing of C. & O. R. R.*
6+85	*R$_T$ 10°00′*	*S. 80°30′ E.*	*S. 80°30′ E.*	
2+90	*R$_T$ 14°30′*	*N. 89°20′ E.*	*N. 89°30′ E.*	*H. B. of Switch*
0		*N. 75°00′ E.*		*Sta. 0 at Benton Sta.*

Checking Angles by the Needle.—In spite of the greatest care, errors in the reading and recording of angles will occur. The best check to such errors is the magnetic needle.

In Fig. 6, we have an example of the use of the needle in *checking* angles. The bearing of the line A B, which corres-

ponds to *A B* in Fig. 5, is N. 75° 00′ E., and is assumed to be correct. The bearing of the line *B C*, as read from the needle is is N. 89° 30′ E. Its *deduced bearing* is obtained as follows: To the bearing of the line *A B*, *viz.*, N. 75° 00′ E., we add the R^r deflection 14° 30′; the sum is 89° 30′, which is recorded in the column headed Ded. Bearing. The deduced bearing, it will be seen, is ten minutes greater than the magnetic bearing read from the needle. Had the deflection angle been recorded L^r instead R^r, the deduced bearing would have been the difference between 75° 00′ and 14° 30′, which is 60° 30′, and would be recorded N. 60° 30′ E. The magnetic bearing being N. 89° 20′ E. would have at once revealed the error. The confusion of the directions R^r and L^r is the commonest source of error in recording deflections, though sometimes a mistake of ten degrees is made in reading the vernier. It is a wise precaution to read both angle and bearing after they are recorded and compare them with the recorded readings.

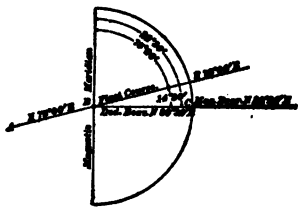


FIG. 6.

TRIANGULATION.

Triangulation is an application of the principles of Trigonometry to the measurement of inaccessible lines and angles. A common occasion for its use is illustrated in Fig. 1, where the line of survey crosses a stream too wide and deep for actual measurement. Set two points *A* and *B* on line, one on each side of the stream. Estimate roughly the distance *A B*. Suppose the estimate is 425 feet. Set another point *C*, making the distance *A C* equal to the *estimated* distance *A B* = 425 feet.

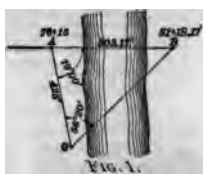


FIG. 1.

Set the transit at A and measure the angle $B\ A\ C =$, say 79° 00′. Next set up at the point C and measure the angle $A\ C\ B =$, say, 56° 20′. The angle $A\ B\ C$ is then determined by subtracting the sum of the angles A and C from 180°; thus, 79° 00′ + 56° 20′ = 135° 20′. 180° 00′ — 135° 20′ = 44° 40′ = the angle $A\ B\ C$. We now have a side and three angles of a triangle given to find the other two sides $A\ B$ and $C\ B$. In Trigonometry, it is demonstrated that, *in any triangle the sines of the angles are proportional to the lengths of the sides opposite to them.* In other words, sin A : sin B :: $B\ C$: $A\ C$; or, sin A : sin C :: $B\ C$: $A\ B$, and sin B : sin C :: $A\ C$: $A\ B$.

Hence, we have sin 44° 40′ : sin 56° 20′ :: 425 : side $A\ B$.

Sin 56° 20′ = .83228;
.83228 × 425 = 353.719;
sin 44° 40′ = .70298;
353.719 ÷ .70298 = 503.17 ft., = side $A\ B$.

Adding this distance to 76 + 15, the station of the point A, we have 81 + 18.17, the station at B.

Another case is the following: Two tangents, $A\ B$ and $C\ D$ (see Fig. 2), which are to be united by a curve, meet at some inaccessible point E. Tangents are the straight portions of a line of railroad. The angle $C\ E\ F$, which the tangents make with each other, and the distances $B\ E$ and $C\ E$ are required. Two points A and B of the tangent $A\ B$, and two points C and D of the tangent $C\ D$ being carefully located, set the transit at

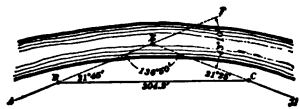


FIG. 2.

B, and backsighting to A, measure the angle $E\ B\ C = 21° 45′$; set up at C, and, backsighting to D, measure the angle $E\ C\ B =$ 21° 25′. Measure the side $B\ C = 304.2$ ft.

Angle $C\ E\ F$ being exterior to the triangle $E\ B\ C$ is equal to the sum of $E\ B\ C$ and $E\ C\ B = 21° 45′ + 21° 25′ = 43° 10′$. The angle $B\ E\ C = 180° - C\ E\ F = 136° 50′$.

From Trigonometry we have

Sin 136° 50′ : sin 21° 45′ :: 304.2 ft. : $C E$;
sin 21° 45′ = .37056 ;
.37056 × 304.2 = 112.724352 ;
sin 136° 50′ = .68412 ;
side $C E$ = 112.724352 ÷ .68412 = 164.77 ft.

Again, we find $B E$ by the following proportion:

Sin 136° 50′ : sin 21° 25′ :: 304.2 : side $B E$;
sin 21° 25′ = .36515 ;
.36515 × 304.2 = 111.07863 ;
sin 136° 50′ = .68412 ;
side $B E$ = 111.07863 ÷ .68412 = 162.36 ft.

A building H, Fig. 3, lies directly in the path of the line $A B$ which must be produced beyond H. Set a plug at B and then turn an angle $D B C = 60°$. Set a plug at C in the line $B C$, at a suitable distance from B, say, 150 feet. Set up at C, and turn an angle $B C D$ = 60°, and set a plug at D, 150 ft. from C. The point D will be in the prolongation of $A B$. Then, set up at D and backsighting to C, turn the angle $C D D' = 120°$. $D D'$ will be the line required, and the distance $B D$ will be 150 feet, since $B C D$ is an equilateral triangle.

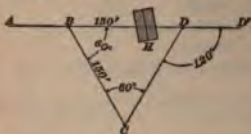


FIG. 3.

$A B$ and $C D$, Fig. 4, are tangents intersecting at some inaccessible point H. The line $A B$ crosses a dock $O P$, too wide for direct measurement, and the wharf $L M$. F is a point on the line $A B$ at the wharf crossing. It is required to find the distance $B H$ and the angle $F H G$. At B, an angle of 103° 30′ is turned to the left and the point E set 217′ from B = to the estimated distance $B F$. Setting up at E, the angle $B E F$ is found to be 39° 00′.

Whence, we find the angle $B F E$ = 180° − (103° 30′ + 39°) = 37° 30′. From Trigonometry we have sin 37° 30′ : sin 39° 00′ :: 217 ft. : side $B F$.

Sin 39° 00′ = .62932 ;
.62932 × 217 = 136.56244 ;
sin 37° 30′ = .60876 ;
side $B F$ = 136.56244 ÷ .60876 = 224.32 ft.

Whence, we find the station of F to be $20 + 17 + 224.33 = 22 + 41.33$. Set up at F and turn an angle $H\ F\ G = 71°\ 00'$ and set up at a point G where the line $C\ D$ prolonged, intersects $F\ G$. Measure the angle $F\ G\ H = 57°\ 50'$ and the side $F\ G = 180.3'$. The angle $F\ H\ G = 180° - (71° + 57°\ 50') = 51°\ 10'$. From Trigonometry we have sin 51° 10′ : sin 57° 50′ :: 180.3 : side $F\ H$.

Sin 57° 50′ = .84650 ;
.84650 × 180.3 = 152.62395 ;
sin 51° 10′ = .77897 ;
side $F\ H = 152.62395 \div .77897 = 195.93$ ft. : whence, we find the station of H to be $24 + 37.26$.

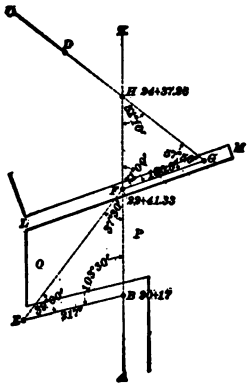


Fig. 4.

CURVES.

Two lines forming an angle of one degree with each other will, at a distance of one hundred feet from the angular point, deflect or diverge 1.745 feet. The *degree of a curve* is determined by the central angle, which is subtended by a chord of 100 feet. Thus, if $B\ O\ G$ (Fig. 1) is 10° and $B\ G$ is 100 feet, $B\ G\ H\ K\ C$ is a 10° curve.

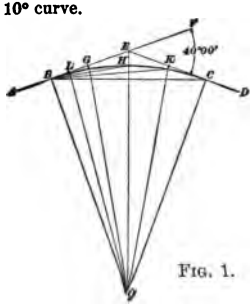


Fig. 1.

The *deflection angle* of a curve is the angle formed at any point of the curve between a tangent and a chord of 100 feet. The deflection angle is therefore *half the degree of the curve.* Thus, if the chord $B\ G$ is 100 feet, the angle $E\ B\ G$ is the deflection angle of the curve $B\ G\ H\ K\ C$ and is half the angle $B\ O\ G$.

Example.—Given the deflection angle E B G = D (Fig. 1), to find the radius B O = R.

SOLUTION.—Draw $O\ L$ perpendicular to $B\ G$. In the right angled triangle $B\ O\ L$, we have $\sin B\ O\ L = \frac{B\ L}{B\ O}$; but $B\ O\ L = E\ B\ G = D$, since $O\ L$ being perpendicular to the chord $B\ G$ bisects the arc $B\ L\ G$. But the angle $D = \frac{1}{2}\ B\ O\ G$: hence, angle $B\ O\ L = D$. $B\ L = 50$ feet and the radius $B\ O = R$. Substituting these values in the given equation, we have $\sin D = \frac{50}{R}$; whence, $R \sin D = 50$, and $R = \frac{50}{\sin D}$.

For curves of from 1° to 10°, the radius may be found by dividing 5,730 ft. (the radius of a 1° curve) by the degree of the curve. The results obtained are sufficiently accurate for all practical purposes. For sharp curves, *i. e.*, for those exceeding 10°, the above formula, *viz.*, $R = \frac{50}{\sin D}$ should be used, especially if the radii are to be used as a basis for further calculation.

Tangent Distances.—When an intersection of tangents has been made and the intersection angle measured, the next question is the degree of curve which is to unite them, which being decided, the next step in order is the location of the points on the tangents where the curve begins and ends. These two points are equally distant from the point of intersection of the tangents, which is called the *P. I.* The point where the curve begins is called the *point of curve*, or the *P. C.*, the point where the curve terminates is called the *point of tangent*, or the *P. T.* The distance of the *P. C.* and *P. T.* from the *P. I.* is called the *tangent distance.*

In Fig. 1, let $A\ B$ and $C\ D$ be tangents intersecting at the point E and forming an angle $C\ E\ F = 40°\ 00'$ with each other. It is decided to unite these tangents by a 10° curve whose radius is 573.7 feet. Call the angle of intersection I, the radius $B\ O$, R, and the tangent distance $B\ E$, T. From geometry we know that $B\ O\ C = C\ E\ F$, hence the angle $B\ O\ E = \frac{1}{2}\ C\ E\ F$. From the right triangle $E\ B\ O$, we have $\tan B\ O\ E = \frac{B\ E}{B\ O}$.

Substituting the above equivalents we have $\tan \frac{1}{2}\ I = \frac{T}{R}$, or $T =$

$R \tan \frac{1}{2} I$. $R = 573.7$; $\frac{1}{2} I = 20°$; $\tan 20° = .36397$. $573.7 \times .36397 = 208.81$ feet. Measure back from the point E on both tangents the distance 208.81 feet to the points B and C. Drive plug flush with the ground at both points and set accurate center points, marked by tacks, in both. Directly opposite each of these plugs drive a stake, called a *guard stake* because it guards or rather indicates where the plug is. The stake at B, if the numbering of the stations runs from B toward C, will be marked *P. C.*, and the stake at C will be marked *P. T.*

To Lay Out a Curve With a Transit.—Having set the tangent points B and C, Fig. 1, set up the transit at B, the *P. C.* Set the vernier at zero and sight to E, the intersection point. Suppose B to be an even or "full station," say 18, and that it has been decided to set stakes at each hundred feet. Let the central angle $B\ O\ G$, measured by the 100 feet chord $B\ G$, be 10°; then, the deflection angle $E\ B\ G$, whose vertex B is in the circumference and subtended by the same chord $B\ G$, will be $\frac{1}{2}\ B\ O\ G$ or 5°. Turn an angle of 5° from B, which in this case will be to the right, measure a full chain 100 feet from B and line in the flag at G; drive a stake at G, which will be marked 19. Turn off an additional 5° making 10° from zero, and at the end of another chain from G, at H, set at a stake marked 20. Continue turning deflections of 5° until 20° or one-half of the intersection angle is reached. This last deflection, if the work has been correctly done, will bring the head chainman to the point of tangent C. It is but rarely that the *P. C.* comes at a full station. When the *P. C.* comes between full stations it is called a *sub-station*, and the chord between it and the next full station is called a *sub-chord*. Had the *P. C.* come at a sub-station, say 17+32, the deflection for the sub-chord of 100 − 32 or 68 feet, the distance to the next station, is found as follows: The deflection for a full station, *i. e.*, 100 feet, is 5° = 300′, and the deflection for 1 foot is $\frac{300'}{100} = 3'$, and for 68 feet the deflection will be $68 \times 3 = 204' = 3°\ 24,'$ which is turned off from zero and a stake set on line, 68 feet from the transit, at station 18. The length of a

curve uniting two given tangents whose intersection is determined, is found as follows:

Suppose $I = 32° 40'$ and that the tangents are to be united by a 6° curve; $32° 40'$ reduced to the decimal form is 32.666°; as each central angle of 6° will subtend a 100 feet chord or one chain, there will be as many such chords or chains as 6 is contained times in 32.666, which is 5.444, that is, there will be 5.444 chains in the curve, or 544.4 feet, which is the required length of the curve. The *P. C.* and *P. T.* having been set and the station of the *P. C.* determined by actual measurement, say $58 + 71$, the station of the *P. T.* is found by adding to $58 + 71$, the station of the *P. C.*, the calculated length of the curve 544.4 feet. $58 + 71 + 544.4 = 64 + 15.4$, the station of the *P. T.*

Tangent and Chord Deflections.—Let *A B* in Fig. 2 be a tangent, and *B C E H* a curve commencing at *B*. Produce the tangent *A B* to the point *D*. The line *C D* is a *tangent deflection*, and is the perpendicular distance from the tangent to the curve. If the chord *B C* be produced to the point *G*, making $C G = B C = C E$, the distance *G E* is a *chord deflection* and is double the tangent deflection *D C*.

Given, the radius *B O* = R, Fig. 2, to find the chord deflection *E G* and the tangent deflection $C D = \frac{1}{2} F E$.

The triangles *O C E* and *C E G* are similar, since both are isosceles, and the angle *G C E* = angle *C O E*. Hence, we have *OC* : *CE* :: *CE* : *EG*. Denoting the chord *C E* by c and the chord deflection *E G* by d, we have, from the above proportion, R : c :: c : d. Therefore, $d = \frac{c^2}{R}$. To find the tangent deflection, draw *C F* to the middle point of *E G*. Then *F E* = the tangent deflection = *D C*. Hence, tangent deflection = one-half the chord deflection, or tangent deflection = $\frac{c^2}{2R}$.

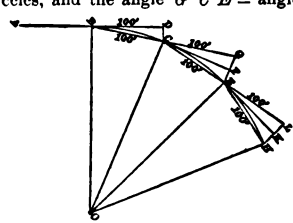


FIG. 2.

Laying Out Curves Without a Transit.—During construction, the engineer is often called upon to restore center stakes on a curve when the transit is not at hand. With the aid of a tape and a few stakes for "lining in", a line can be run closely approximating to the true one, by applying the principle just demonstrated.

A practical application of this principle is shown in Fig. 3, in which $A\,B$ is a tangent, B the *P. C.* of a 4° curve R*T*. The chord deflection of a 4° curve for 100 feet chord, is 6.98 ft. The tangent deflection = ½ the chord deflection is 3.49 ft. Let B = Sta. 8 + 25; a stake at each full station on the curve is required. Restore the stakes at A and B, which determine the *P. C.*, and give the direction of the tangent $A\,B$. The distance from the *P. C.* to the next full station C is 75 ft. or .75 of a full station; $75^2 \times 3.49 = .562 \times 3.49 = 1.96$ ft., the tangent deflection at C. The point C is found by measuring 75 ft. from B and setting a stake at C in line with a stake at B, the *P. C.*, and a point on the tangent $A\,B$ as A. With a tape, measure the distance 1.96 ft. from C at right angles to $B\,C$, and drive a stake at that point F, which will be station 9. Measure 100 ft. from F and set at a point at D in the line $B\,F$. By previous calculation, we know that the chord deflection for 100 ft. is 6.98 ft. Measure the distance 6.98 ft. at right angles to the line $F\,D$ and drive a stake at G, which will be station 10. In like manner set the remaining station 11, which is previously known to be the *P. T.* Although the chord deflection $D\,G$ is not theoretically at right angles to $F\,D$, yet $D\,G$ is so small compared with $F\,D$ that for curves of ordinary degree the offset is made at right angles.

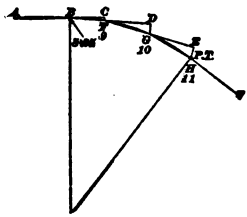


FIG. 3.

To Determine Degree of Curve by Measuring a Middle Ordinate.—In track work, it is often necessary to know the degree of a curve when no transit is available for measuring it. The degree can be found by measuring the middle ordinate of

convenient chord, and multiplying its length by 8, which will give the chord deflection for that curve.

Let *A B* in Fig. 4, be a 50 feet chord, measured on the track, and let the middle ordinate *a b*, be .44 ft. $.44 \times 8 = 3.52$ = chord deflection for 50 ft., which, expressed in decimal part of a full station, is .5; $.5^2 = .25$. The chord deflection for 100 ft. multiplied by .25 = the chord deflection for 50 ft., which we know by calculation to be 3.52 ft. Hence, $3.52 \div .25 = 14.08$ ft., the chord deflection for 100 ft., which is divided by 1.745, the chord deflection for a 1° curve, gives a quotient of 8.07 nearly. The inference is that the curve is 8°.

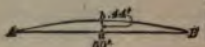


FIG. 4.

How to Keep Transit Notes.—A good form for location notes is the following:

June 30, 1894

Station.	Deflection.	Tot. Angle.	Mag. Bearing	Ded. Bearing	Remarks.	
9						
8						
7						
6+95	4°24' P.T.	15°00'	N. 35°30' E.	N. 35°15' E.		
6+50	4°00'					
6	3°00'					"To Construction of Railway"
5+00	3°00'				5+00	
5	1°00'				5+50	
4+50	2°30'	6°12'				
4	1°20'				Int. Angle=15°00'	4° Curve 2°
3+50	0°30'				T=188.81 ft.	Def. Angle for 50 ft=8°
3+20	P.C. 4° R.				P.C=3+20	Def. Angle for 1 ft=1.2°
3					Length of Curve=375 ft.	
2					P.T=6+95	
1						
0			N. 20°30' E.	N. 20°15' E.		

In the first column the station numbers are recorded. In the second column are recorded the deflections with the abbreviations *P. C.* and *P. T.* together with the degree of curve and the abbreviation Rt or Lt, according as the line curves to the right or left. At each transit point on the curve, the total or central angle from the *P. C.* to that point is calculated and recorded in the third column. This total angle is double the deflection angle between the *P. C.* and the transit point. In the above

notes, there is but one intermediate transit point between the *P. C.* and *P. T.* The deflection from *P. C.* at Sta. 3+20 to the intermediate transit point at Sta. 4+50, is 2° 36′. The total angle is double this deflection, or 5° 12′, which is recorded on the same line in the third column. The record of total angles at once indicates the stations at which transit points are placed. The total angle at the *P. T.* will be the same as the angle of intersection, if the work is correct. When the curve is finished, the transit is set up at the *P. T.* and the bearing at the forward tangent taken, which affords an additional check upon the previous calculations. The magnetic bearing is recorded in the fourth column and the deduced or calculated bearing is recorded in the fifth column.

Leveling.

Examples in Direct Leveling.—The principles of direct leveling are illustrated in Fig. 1.

Let *A* be the starting point, which has a known elevation of 20 ft. The instrument is set at *B*, leveled up and sighted to a rod held at *A*. The target being set, the reading, 8.42 ft., called a backsight, is the distance which the point where the line of sight cuts the rod is above the point *A*, and is to be added to the elevation of the point *A*. $20.00 + 8.42 = 28.42$, is called the height of instrument and designated *H. I.* The instrument being turned in the opposite direction, a point *C* is chosen, which must be below the line of sight. This point is called a turning point and is designated by the abbreviation *T. P.* Drive a peg at *C*, or take for a turning point a point of rock or some other permanent object upon which the rod is held. The reading at this point is a foresight, and is to be subtracted from the height of the instrument at *B* to find the elevation of the point at *C*.

Let the rod reading be 1.20 ft. As this reading is a foresight, it must be subtracted from 28.42, the height of instrument at *B*; $28.42 - 1.20 = 27.22$ ft., the elevation of the point *C*. The leveler carries the instrument to *D*, which should be of such a height above *C*, that when leveled up, the line of sight will cut the rod *near the top*. The backsight to C gives a reading of 11.58 ft., *which* added to 27.22 ft., the elevation of C, gives 38.78 ft., the

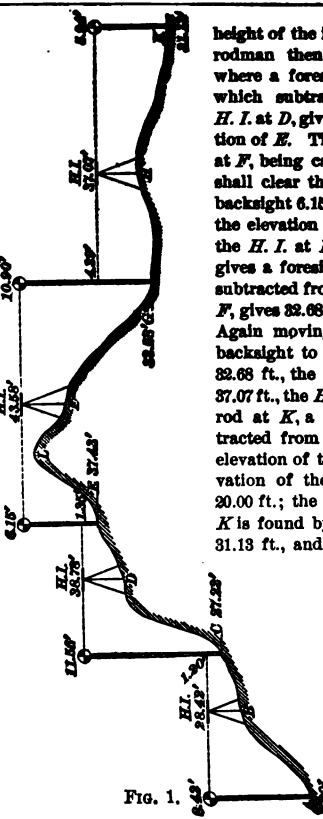


FIG. 1.

height of the instrument at *D*. The rodman then goes to *E*, a point where a foresight reading is 1.25, which subtracted from 38.78, the *H. I.* at *D*, gives 37.43 ft., the elevation of *E*. The level is then set up at *F*, being careful that line of sight shall clear the hill at the *L*. The backsight 6.15 ft. added to 37.43 ft., the elevation of *E*, gives 43.58 ft., the *H. I.* at *F*. The rod held at *G* gives a foresight of 10.90 ft., which subtracted from 43.58 ft., the *H. I.* at *F*, gives 32.68 ft., the elevation at *G*. Again moving the level to *H*, the backsight to *G* of 4.39 ft. added to 32.68 ft., the elevation of *G*, gives 37.07 ft., the *H. I.* at *H*. Holding the rod at *K*, a foresight of 5.94 subtracted from 37.07 gives 31.13, the elevation of the point *K*. The elevation of the starting point *A* is 20.00 ft.; the elevation of the point *K* is found by direct leveling to be 31.13 ft., and the difference in the elevations of *A* and *K* is 31.13 — 20.00 = 11.13 ft.; that is, the point *K* is 11.13 ft. higher than the point *A*.

Turning points previously mentioned are the points where backsights and foresights are taken. The back-*sights* are plus (+) readings, and are to be added; the fore-

sights are minus (—) readings, and are to be subtracted. A point for a foresight having been determined, the rodman drives a peg firmly in the ground and holds the rod upon it. After the instrument is moved, set up and a backsight taken, the peg is pulled up and carried in the pocket until another turning point is called for. Turning points should be taken at about equal distances from the instrument in order to equalize any small errors in adjustment. In smooth country an ordinary level will permit of sights of from 300 to 500 feet.

How to Keep Level Notes.—Forms for keeping level notes are various. One of the best forms in general use is shown in the diagram on the following page.

The distinguishing feature of this form of level notes is a single column for all rod readings. The backsights being additive and the foresights subtractive readings, they are distinguished from other rod readings by the characteristic signs + (plus) and — (minus). The turning points, whose foresight readings are —, are further designated by the abbreviation *T. P.*

How to Check Level Notes.—There is one method of checking level notes which is in universal use. It provides for checking the elevations of turning points and heights of instrument only, which is sufficient, as all other elevations are deduced from them. The method depends upon the fact that all backsights are additive or + quantities, and all foresights are subtractive or — quantities. The level notes given in the diagram on the following page are checked as follows: The elevation of the bench mark at station 0 is 100.00 feet, to which all backsights or + readings are to be added and from this sum all foresights or — readings are to be subtracted.

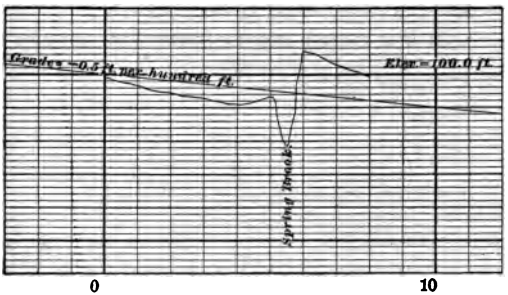
The sum of the + readings or backsights together with the elevation of the bench mark at 0 is 122.59. The sum of the — readings or foresights is 24.27, and the difference 98.32 feet is the elevation of the turning point last taken. As soon as a page of level notes is filled, the leveler should check them, placing a check mark ✓ at the last height of instrument or elevation checked. When the work of staking out or cross-sectioning is being done, the levels should be checked at each *bench mark* on the line. At the close of each day's work, the *leveler must* check on the nearest bench mark.

+	—
Thus 100.00	10.22
5.61	2.52
5.41	11.53
11.57	24.27
122.59	
24.27	
98.32	

1. Station.	Rod Reading.	Ht. Instrument.	Elevation.	Grade.	Cut.	Fill.	Remarks.	2. Date.
B. M.	+ 5.61	105.61	100.00				On Root of White Oak.	St'mp 10' L. Sta. 0.
0	6.1		99.5					
1	7.3		98.3					
2	8.4		97.2					
3	9.2		96.4					
T. P.	— 10.22		95.39					
	+ 5.41	100.80						
4	6.3		94.5					
5	4.2		96.6					
5 + 50	11.5		89.3				Spring Brook.	
T. P.	— 2.52		98.28					
	+ 11.57	109.85						
6	6.2		103.6					
7	8.5		101.3					
8	10.1		99.7					
T. P.	— 11.53		98.32					

Profiles.—A profile represents a vertical section of the line of survey. In it all abrupt changes in elevation are clearly outlined. Vertical and horizontal measurements are usually represented by different scales, to render irregularities of surface more distinct through exaggeration. For railroad work profiles are commonly made to the following scales: *viz.*, horizontal, 400 feet = 1 inch; vertical, 20 feet = 1 inch.

A section of profile paper is shown in the following diagram. Every fifth horizontal line and every tenth vertical line are heavy. By the aid of these heavy lines, distances and elevations are quickly and correctly estimated and the work of platting greatly facilitated. The level notes given in the preceding diagram are platted in the accompanying section. The elevation of



some horizontal line is assumed. This elevation is, of course referred to the datum line, and is the base from which the other elevations are estimated. Every tenth station number is written at the bottom of the sheet under the heavy vertical lines. The profile is first platted in pencil and then inked in black.

Grade Lines.—The principal use of a profile is to enable the engineer to establish a *grade line*; *i. e.*, a line showing the relative proportion of excavation and embankment in the complete work. The *rate* of a grade line is measured by the vertical rise or fall in each hundred feet of its length, and is designated by the term per cent. Thus, a grade line which rises or falls 1 foot

in each hundred feet of its length is called an ascending or descending 1 per cent grade, and written + 1.0 or — 1.0 per hundred. A rise or fall of one-half foot in each hundred feet is called a five tenths per cent. grade, and written + 0.5 or — 0.5 per hundred. The grade line having been decided upon, it is drawn in red ink.

EXAMPLE.—The elevation of station 20 is 140.0 feet; between stations 20 and 100 there is an ascending grade of 0.75 per cent.; what is the elevation of the grade at station 71 ?

SOLUTION.—To obtain the elevation of the grade at station 71, we add to the elevation of the grade at station 20, 140 feet, the total rise in grade between stations 20 and 71. Accordingly, 71 — 20 = 51 ; 0.75 foot × 51 = 38.25 feet ; 140 feet + 38.25 feet = 178.25 feet, the elevation of grade at station 71.

RADII AND CHORD AND TANGENT DEFLECTIONS.

The formulas used in the computation of the following tables are as follows :

For Radii, $R = \frac{50}{\sin D}$.

For Chord Deflection, $d = \frac{c^2}{R}$.

For Tangent Deflections, tan deflection $= \frac{c^2}{2R}$.

TABLE OF RADII AND DEFLECTIONS.

Degree.	Radii.	Chord Deflection.	Tangent Deflection.	Degree.	Radii.	Chord Deflection.	Tangent Deflection.
° ′				° ′			
0 5	68754.94	.145	.073	1 0	5729.65	1.745	.873
10	34377.48	.291	.145	5	5288.92	1.891	.945
15	22918.33	.436	.218	10	4911.15	2.036	1.018
20	17188.76	.582	.291	15	4583.75	2.182	1.091
25	13751.02	.727	.364	20	4297.28	2.327	1.164
30	11459.19	.873	.436	25	4044.51	2.472	1.236
35	9822.18	1.018	.509	30	3819.83	2.618	1.309
40	8594.41	1.164	.582	35	3618.80	2.763	1.382
45	7639.49	1.309	.654	40	3437.87	2.909	1.454
50	6875.55	1.454	.727	45	3274.17	3.054	1.527
55	6250.51	1.600	.800	50	3125.36	3.2000	1.600

TABLE OF RADII AND DEFLECTIONS—*continued.*

Degree.	Radii.	Chord Deflection.	Tangent Deflection.	Degree.	Radii.	Chord Deflection.	Tangent Deflection.
° ′				° ′			
1 55	2989.48	3.345	1.673	5 15	1091.73	9.160	4.580
				20	1074.68	9.305	4.653
2 0	2864.93	3.490	1.745	25	1058.16	9.450	4.725
5	2750.35	3.636	1.818	30	1042.14	9.596	4.798
10	2644.58	3.781	1.891	35	1026.60	9.741	4.870
15	2546.64	3.927	1.963	40	1011.51	9.886	4.943
20	2455.70	4.072	2.036	45	996.87	10.031	5.016
25	2371.04	4.218	2.109	50	982.64	10.177	5.088
30	2292.01	4.363	2.181	55	968.81	10.322	5.161
35	2218.09	4.508	2.254				
40	2148.79	4.654	2.327	6 0	955.37	10.467	5.234
45	2083.68	4.799	2.400	5	942.29	10.612	5.306
50	2022.41	4.945	2.472	10	929.57	10.758	5.379
55	1964.64	5.090	2.545	15	917.19	10.903	5.451
				20	905.13	11.048	5.524
3 0	1910.08	5.235	2.618	25	893.39	11.193	5.597
5	1858.47	5.381	2.690	30	881.95	11.339	5 669
10	1809.57	5.526	2.763	35	870.79	11.484	5.742
15	1763.18	5.672	2.836	40	859.92	11.629	5.814
20	1719.12	5.817	2.908	45	849.32	11.774	5.887
25	1677.20	5.962	2.981	50	838.97	11.919	5.960
30	1637.28	6.108	3.054	55	828.88	12.065	6.032
35	1599.21	6.253	3.127				
40	1562.88	6.398	3.199	7 0	819.02	12.210	6.105
45	1528.16	6.544	3.272	5	809.40	12.355	6.177
50	1494.95	6.689	3.345	10	800.00	12.500	6.250
55	1463.16	6.835	3.417	15	790.81	12.645	6.323
				20	781.84	12.790	6.395
4 0	1432.69	6.980	3.490	25	773.07	12.936	6.468
5	1403.46	7.125	3.563	30	764.49	13.081	6.540
10	1375.40	7.271	3.635	35	756.10	13.226	6.613
15	1348.45	7.416	3.708	40	747.89	13.371	6.685
20	1322.53	7.561	3.781	45	739.86	13.516	6.758
25	1297.58	7.707	3.853	50	732.01	13.661	6.831
30	1273.57	7.852	3.926	55	724.31	13.806	6.903
35	1250.42	7.997	3.999				
40	1228.11	8.143	4.071	8 0	716.78	13.951	6.976
45	1206.57	8.288	4.144	5	709.40	14.096	7.048
50	1185.78	8.433	4.217	10	702.18	14.241	7.121
55	1165.70	8.579	4.289	15	695.09	14.387	7.193
5 0	1146.28	8.724	4.362	20	688.16	14.532	7.266
5	1127.50	8.869	4.435	25	681.35	14.677	7.338
10	1109.33	9.014	4.507	30	674.69	14.822	7.411

TABLE OF RADII AND DEFLECTIONS—*continued.*

Degree.	Radii.	Chord Deflection.	Tangent Deflection.	Degree.	Radii.	Chord Deflection.	Tangent Deflection.
° ′				° ′			
8 35	668.15	14.967	7.483	13 40	420.23	23.796	11.898
40	661.74	15.112	7.556	50	415.19	24.085	12.043
45	655.45	15.257	7.628				
50	649.27	15.402	7.701	14 0	410.28	24.374	12.187
55	643.22	15.547	7.773	10	405.47	24.663	12.331
				20	400.78	24.951	12.476
9 0	637.27	15.692	7.846	30	396.20	25.240	12.620
5	631.44	15.837	7.918	40	391.72	25.528	12.764
10	625.71	15.982	7.991	50	387.34	25.817	12.908
15	620.09	16.127	8.063				
20	614.56	16.272	8.136	15 0	383.06	26.105	13.053
25	609.14	16.417	8.208	10	378.88	26.394	13.197
30	603.80	16.562	8.281	20	374.79	26.682	13.341
35	598.57	16.707	8.353	30	370.78	26.970	13.485
40	593.42	16.852	8.426	40	366.86	27.258	13.629
45	588.36	16.996	8.498	50	363.02	27.547	13.773
50	583.38	17.141	8.571	16 0	359.26	27.835	13.917
55	578.49	17.286	8.643	10	355.59	28.123	14.061
				20	351.98	28.411	14.205
10 0	573.69	17.431	8.716	30	348.45	28.699	14.349
10	564.31	17.721	8.860	40	344.99	28.986	14.493
20	555.23	18.011	9.005	50	341.60	29.274	14.637
30	546.44	18.300	9.150				
40	537.92	18.590	9.295	17 0	338.27	29.562	14.781
50	529.67	18.880	9.440	10	335.01	29.850	14.925
				20	331.82	30.137	15.069
11 0	521.67	19.169	9.585	30	328.68	30.425	15.212
10	513.91	19.459	9.729	40	325.60	30.712	15.356
20	506.38	19.748	9.874	50	322.59	31.000	15.500
30	499.06	20.038	10.019				
40	491.96	20.327	10.164	18 0	319.62	31.287	15.643
50	485.05	20.616	10.308	10	316.71	31.574	15.787
				20	313.86	31.861	15.931
12 0	478.34	20.906	10.453	30	311.06	32.149	16.074
10	471.81	21.195	10.597	40	308.30	32.436	16.218
20	465.46	21.484	10.742	50	305.60	32.723	16.361
30	459.28	21.773	10.887				
40	453.26	22.063	11.031	19 0	302.94	33.010	16.505
50	447.40	22.352	11.176	10	300.33	33.296	16.648
				20	297.77	33.583	16.792
13 0	441.68	22.641	11.320	30	295.25	31.870	16.935
10	436.12	22.930	11.465	40	292.77	34.157	17.078
20	430.69	23.219	11.609	50	290.33	34.443	17.222
30	425.40	23.507	11.754	20 0	287.94	34.730	17.365

RETAINING WALLS.

On the Theory of Retaining Walls.—Let *a b d c*, Fig. 1, be a retaining wall with battered face and vertical back. The top *b e* of the backing is level with the top of the wall. Let *d e* represent the natural slope of the material composing the filling, *viz.*, 1½ horizontal to 1 vertical, which is the average of materials used for back filling.

It is assumed that the wall *a b d c* is heavy enough to resist sliding along its base and that it can fail only by overturning, *i. e.*, rotating about its toe *c*. Now if the angle *o d e* between the vertical line *o d* drawn from the inner bottom edge of the wall and the natural slope *d e* be bisected by the line *d f*, the *angle o d f* is called the *angle* and the *line d f* the *slope of maximum pressure*. The triangular prism of earth *o d f* is called the *prism of maximum pressure* because, if considered as a wedge acting against the back of the wall, it would exert a greater pressure against it than would the entire triangle *o d e* of earth considered as a single wedge. For though the last is more than double the weight of the former, yet it receives much greater support from the underlying earth. It has been proved by experiment that if the triangle of earth *o d e* be divided by any line *d f* into wedges, the wedge that will press most against the wall is that formed when the line *d f* divides the angle *o d e* into two equal parts.

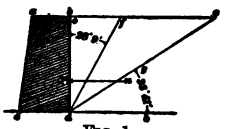


FIG. 1.

The angle *o d h* is formed by the vertical *o d* and the horizontal *d h* is 90°. The angle of natural slope *h d e* is 33° 41′; hence, the angle *o d f* of maximum pressure is equal to 90° — 33° 41′ ÷ 2 = 28° 09′.

In making calculations, only *one foot* of the length of *wall* and of the *backing* is taken, so all that is necessary is to take the area of the section of the wall and backing. The material composing the backing is supposed to be perfectly dry and possessing no cohesive power which is practically true of pure sand.

If we conceive the wall *a b d c*, Fig. 1. to be suddenly removed, the triangle *b d f* of sand included between the line of

maximum pressure *d f* and the vertical back *b d* of the wall would slide downward impelled by a force *n P*; acting in a direction *n P* at right angles to the side *b d* of the triangle, *i.e.*, at right angles to the vertical back *b d* of the wall; *its center of force* being at *P* distant ⅓ way between *b* and *d* measured from the bottom of the wall *d*. The amount of this force is equal to the following:

$$\text{Perpendicular pressure } n\,P = \frac{\text{weight of triangle of earth } b\,d\,f \times o\,f}{\text{vertical depth } o\,d}.$$

This formula not only applies to walls with vertical backs, as in Fig. 1, but to those with inclined backs as in Fig. 2, for inclinations as high as 6 inches horizontal to 1 foot vertical, which is rarely met with and never exceeded.

Friction Caused by Pressure of Backing.—If all the backing material contained between the line of natural slope and the back of the wall were unconfined, it would slide, producing *motion*, but confined by the retaining wall, the force is converted into *pressure* of earth against the back of the wall, resisted by the *friction* between the compressed earth and the wall.

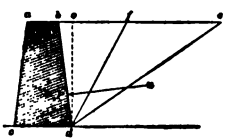


FIG. 2.

If the wall were to begin to overturn about its toe *c* (Figs. 1 and 2) as a fulcrum, its back *b d* would rise, producing friction against the backing. So long as the wall does not move, the friction of the backing acts constantly, and must, therefore, be one of the forces which *prevent* overturning. We ascertain the amount and effect of this friction as follows: Let *a b d c*, Fig. 3, be a retaining wall and let *n P* represent to some *scale* the perpendicular pressure against the back of the wall calculated by the preceding formula, *viz.*, perpendicular pressure =

$$n\,P = \frac{\text{weight of triangle } d\,b\,f \times o\,f}{\text{vertical depth } o\,d}.$$

Make the angle n P h equal to the angle of wall friction, ***viz., that at which a*** plane of masonry must be inclined to the hori-

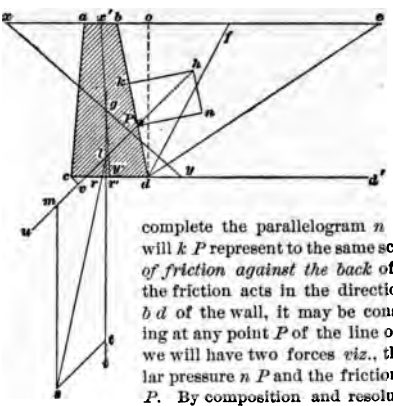


FIG. 3.

zontal in order that dry sand and earth may slide freely over it, and taken at 38° 41′. Draw n h perpendicular to n P and complete the parallelogram n h k P. Then, will k P represent to the same scale the *amount of friction against the back* of the wall. As the friction acts in the direction of the back b d of the wall, it may be considered as acting at any point P of the line of the back and we will have two forces *viz.*, the perpendicular pressure n P and the friction k P acting at P. By composition and resolution of forces, the diagonal h P measured to the same scale will give us the amount of their resultant, which is approximately the *single theoretical force* both in *amount* and *direction* which the wall has to resist. This force includes the wall friction. The force h P is always equal to the perpendicular force n P, divided by the cosine of the angle of wall friction. The cosine of the angle of wall friction is .832 and the value of the force h P may be expressed in the following formula :

Approximate theoretical pressure

$$h\,P = \frac{\text{weight of triangle } b\,d\,f \times o\,f}{\text{vertical height } o\,d \times .832}.$$

when the back of the wall does not incline forward more than 6 inches horizontal to 1 foot vertical, equal to an angle of about 26° 34′, the following formula by Trautwine is used, *viz.:*

Approximate theoretical pressure

$$h\,P = \text{weight of triangle } b\,d\,f \times .643,$$

which includes friction of earth against the back of the wall.

To Find the Overturning and Resisting Forces.—To find the *overturning tendency of the earth pressure* and the resistance of

the wall against being overturned about its toe c, as a fulcrum, see Fig. 3. Find the center of gravity *g* of the wall, and through *g* draw the vertical line *g i*. Produce the line of pressure *h P* and draw *c v* at right angles to this line. To any convenient scale, lay off *l t* equal to the weight of the wall and to the same scale *l m* equal to the pressure *h P*. Complete the parallelogram *l m s t*. The diagonal *l s* will be the resultant of the pressure and the weight of the wall. The stability of the wall will be the greater as the distance *c r*, from the toe to the point where the resultant *l s* cuts the base, increases. To insure stability, *c r* must be greater than $\frac{1}{3}$ *c d*.

The pressure *h P*, if multiplied by its leverage *c v* will give the moment of the pressure about *c* and the weight of the wall *l t* multiplied by its leverage *c r'* will give the moment of the wall. The wall is secure against overturning in proportion as its moment exceeds that of the pressure.

For example, let the height of the wall *a b d c*, in Fig. 3, be 9 ft.; the thickness at the base *c d*, 4.5 ft. and at the top *a b*, 2 ft. and the batter of *a c* be 1 in. to the ft. The triangle of earth *b d f* has a base *b f* = 6.57 ft. and altitude *d o* = 9 ft. Taking the section as 1 ft. in thickness, we have the contents equal to 6.57 × 9 ÷ 2 = 29.56 cu. ft. Assuming the material to weigh 120 lb. per cu. ft., the weight of the triangle *b d f* is 29.56 × 120 = 3,547 lb., *o f* = 4.81 ft. 3,547 × 4.81 = 17,061. 17,061 ÷ *o d*, = 1,895.7 lb, = the perpendicular pressure *n P*. Lay off on a line perpendicular to the back of the wall at *P*, to a scale of 2,000 lb. = 1 in., *n P* = 1,895.7 ÷ 2,000 = 0.948 in., the perpendicular pressure. Draw *P h*, making the angle *n P h* = 33° 41′. Draw *n h* intersecting *h P* in *h*; then, will *n h* to the same scale equal the friction of the earth against the back of the wall. Completing this parallelogram, *n h k P*, the diagonal *h P* = 1.139 in., which to a scale of 2,000 lb. = 1 in., amounts to 2,278 lb., and is the resultant of the pressure and the friction.

Produce the resultant *h P* to *u*. We next find the center of gravity *g* of the wall *a b d c*. The section of the wall is a trapezoid and the center of gravity *g* is readily found as follows: *Produce the* upper base of the section to *x*, making *a x* = *c d* = *4.5 feet*. Produce the lower base in the opposite direction to *y*

making $d\,y = a\,b = 2$ ft. Join x and y. Find the middle points x' and y' of the upper and lower bases of the section. Join these points. The intersection g of the lines $x\,y$ and $x'\,y'$ is the center of gravity of the trapezoid $a\,b\,d\,c$.

The volume of the section of wall $a\,b\,d\,c$ is readily found. The sum of top and bottom widths $= 2.0 + 4.5 = 6.5$ ft. $6.5 \div 2 = 3.25$ ft. $3.25 \times 9 = 29.25$ cu. ft. $29.25 \times 154 = 4{,}504$ lb., (the weight per cubic foot of good mortar rubble $= 154$ lb.) $=$ the weight of the section $a\,b\,d\,c$. Draw through g a vertical line $g\,i$ and lay off in it to a scale of 2,000 lb. to the inch from the point i, where the line of gravity intersects the prolongation of the line of pressure $h\,P$, the length $l\,t$ equal to 4,504 lb., the weight of the wall. Lay off from l on the prolongation of $h\,P$, $l\,m$ equal to 2,278 lb. to the same scale. Complete the parallelogram $l\,m\,s\,t$. The diagonal $l\,s$ represents the resultant of the pressure and of the weight of the wall. The distance $c\,r$ from the toe c to the intersection of the resultant $l\,s$ with the base $c\,d$ is more than $\frac{1}{6}$ of the width of the base which insures ample stability.

Pressure of the Backing on Surcharged Walls.—In Fig. 4, the surcharge of backing $m\,b\,o$ slopes from b at its natural slope and attains its maximum pressure where the slope of maximum pressure $d\,k$ intersects the natural slope $b\,m$ at f. Any additional height of surcharge does not increase this pressure. If the surcharge slopes from a, as shown by the line $a\,p$, or from any point between a and b, then the slope of maximum pressure must be extended, intersecting the slope from a in the point k. The prism of maximum pressure will then be $d\,i\,k$. The triangle of earth $a\,b\,i$ on the top of the wall exerts no *pressure against* the back of the wall, but adds to its stability.

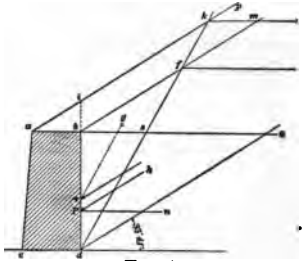


FIG. 4.

Having found the weight of the triangle $b\ d\ f$, we have

approximate pressure = weight of triangle $b\ d\ f \times .643$,

which includes the pressure of the backing and the friction of the earth against the back of the wall.

Draw $P\ n$ perpendicular to the back of the wall and draw $h\ P$ making the angle $n\ P\ h = 33° 41'$, the angle of wall friction. Then, $h\ P$ will be the direction of the pressure. The point of application of this pressure will not always be at P, $\frac{1}{3}$ of the height of $b\ d$ measured from d, but above P as at r where a line drawn from the center of gravity g of the *prism of maximum pressure* $d\ i\ k$ (omitting any earth resting directly upon the top of the wall) and parallel to the line $d\ k$ of maximum pressure, cuts the back $b\ d$ of the wall. The center of pressure P will be at $\frac{1}{3}$ the height of the wall when the sustained earth $d\ b\ s$ or $d\ b\ f$ forms a *complete triangle* one of whose angles is at b the *inner top edge* of the wall. For all other surcharges, the point of pressure will be *above P*.

TUNNEL SECTIONS.

Tunnel sections vary somewhat, according to the material to be excavated, but the general form and dimensions are much the same.

Section of Double Track Tunnel.

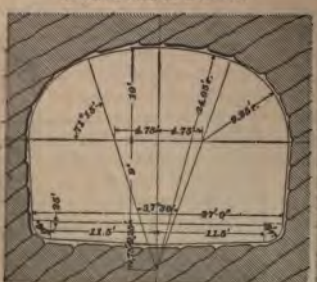


FIG. 1.

Section of Single Track Tunnel.

FIG. 2.

The general dimensions are as follows: For double track, *from 22 to 27* feet wide and from 21 to 24 feet high, and for single

track, from 14 to 16 feet wide and from 17 to 20 feet high. See Figs. 1 and 2.

In seamy or rotten rock the section is sufficiently enlarged to receive a lining of substantial rubble or brick masonry laid in good cement mortar. When the material has not sufficient consistency to sustain itself until the masonry lining is built, resort is had to timbering, which furnishes the necessary support.

CALCULATION OF EARTHWORK.

In calculating the quantity of material in excavation and embankment, two general methods are used, namely, what is known as the *end area* and the *prismoidal formula*.

Calculation by the end area method consists in multiplying the mean or average area in square feet of two consecutive sections by the distance in feet between them.

Thus, let A represent the area in square feet of one section; B, the area in square feet of the next section; C, the number of feet between the sections, and D, the total number of cubic feet in the prismoid lying between these sections. Then,

$$D = \frac{A + B}{2} \times C, \text{ approximately.}$$

The distance between sections should not be more than 100 feet, and should be less if the surface of the ground is irregular.

A more accurate result is obtained by the use of the prismoidal formula. In applying the prismoidal formula to the calculation of cubic contents it is requisite to know the middle cross-section between each two that are measured on the ground. The dimensions of this middle section are the mean of the dimensions of the end sections.

Calling one of the given sections A, the other B, the average or mean section M, the distance between the sections L, and the required contents S, we have, by the prismoidal formula,

$$S = \frac{L}{6}(A + 4M + B).$$

EXAMPLE.—Two sections are represented by Figs. 1 and 2, and are denoted by the letters *A* and *B*. The perpendicular distance

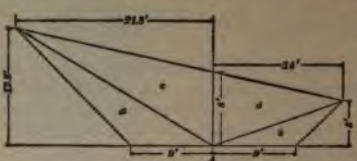


Fig. 1.

between them is 50 feet. It is required to find the cubical contents of the prismoid.

Solution.—The section given in Fig. 1 is composed of the four triangles, a, b, c and d. The triangles a and b have equal bases of 9 feet, the half width of the roadway; hence, if we take half the sum of their altitudes and multiply it by the common base we shall have the sum of the areas of the triangles a and b.

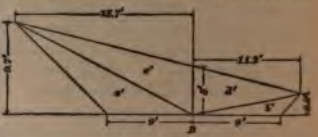


Fig. 2.

The triangles c and d have a common base 8 feet, the center cut of the section, and if we take the half sum of the side distances and multiply it by 8 feet, we shall obtain the areas of the triangles c and d. Taking the dimensions of section A given in Fig. 1, we have

$$\text{Areas of triangles } a+b = \frac{12.8+5}{2} \times 9 = 80.1 \text{ sq. ft.}$$

$$\text{" " " } c+d = \frac{21.8+14}{2} \times 8 = 143.2 \text{ " "}$$

$$\text{Total area of section } A = 223.3 \text{ " "}$$

Taking the dimensions of the section B given in Fig. 2, we have

$$\text{Areas of triangles } a'+b' = \frac{9.7+2.2}{2} \times 9 = 53.55 \text{ sq. ft.}$$

$$\text{" " " } c'+d' = \frac{18.7+11.2}{2} \times 5 = 74.75 \text{ " "}$$

$$\text{Total area of section } B = 128.3 \text{ " "}$$

In applying the prismoidal formula we calculate the area of a *tion midway* between the given sections, and for its dimen-

sions we take the mean of the dimensions of the given sections. These dimensions will be as follows:

Center cut, $\frac{8+5}{2} = 6.5$ ft.

Right side distance, $\frac{14+11.2}{8} = 12.6$ ft.

Left side distance, $\frac{21.8+18.7}{2} = 20.25$ ft.

With dimensions thus found, construct the section M shown in Fig. 3.

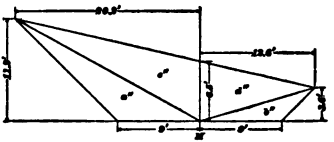


FIG. 3.

The area of section M is computed by the same method as that used with sections A and B in Figs. 1 and 2, and is as follows:

Area of triangles $a'' + b'' = \frac{11.2+3.6}{2} \times 9 = 66.6$ sq. ft.

" " " $c'' + d'' = \frac{20.2+12.6}{2} \times 6.5 = 106.6$ " "

Total area of section $M = 173.2$ " "

Denoting the distance between the sections by L, and the cubical contents of the prismoid by S, we have, by substituting in the prismoidal formula,

$$S = \frac{L}{6}(A + 4M + B).$$

$$S = \frac{50}{6}(223.3 + 4 \times 173.2 + 128.3) = 8{,}703 \text{ cu. ft.} = 322.3 \text{ cu. yd.}$$

TRACK WORK.

Curving Rails.—When laying track on curves, in order to have a smooth line, the rails themselves must conform to the curve of the center line. To accomplish this, the rails must be curved. The curving should be done with a rail bender or with a lever, preferably with the former.

To guide those in charge of this work, a table of middle and quarter ordinates for a 30-foot rail for all degrees of curve should be prepared.

The following table of middle ordinates for curving rails is calculated by using the formula

$$m = \frac{c^2}{8R},$$

in which m is the middle ordinate; c, the chord, assumed to be of the same length as the rail, and R, the radius of the curve.

The results obtained from this formula are not theoretically correct, yet the error is so small that it may be ignored in practical work.

In curving rails, the ordinate is measured by stretching a cord from end to end of the rail against the gauge side, as shown in Fig. 1. Suppose the rail $A\ B$ is 30 feet in length, and the curve 8°. Then, by the previous problem, the middle ordinate at a should be $1\frac{7}{8}$ inches. To insure a uniform curve to the rails, the ordinates at the quarters b and b' should be tested. In all cases the quarter ordinates should be three-quarters of the middle ordinate. In Fig. 1, if the rail has been properly curved, the quarter ordinates at b and b' will be $\frac{3}{4} \times 1\frac{7}{8}$ in. $= 1\frac{21}{32}$, say $1\frac{5}{8}$ in.

FIG. 1.

MIDDLE ORDINATES FOR CURVING RAILS.

Degree of Curve.	*Lengths of Rails.*					
	30 ft.	*28 ft.*	*26 ft.*	*24 ft.*	*22 ft.*	*20 ft.*
	in.	*in.*	*in.*	*in.*	*in.*	*in.*
1	0 1/4	0 3/16	0 3/16	0 3/16	0 1/8	0 1/8
2	0 1/2	0 7/16	0 3/8	0 5/16	0 1/4	0 3/16
3	0 3/4	0 5/8	0 9/16	0 7/16	0 3/8	0 5/16
4	0 15/16	0 13/16	0 11/16	0 5/8	0 1/2	0 7/16
5	1 3/16	1 1/16	0 15/16	0 3/4	0 5/8	0 9/16
6	1 7/16	1 1/4	1 1/16	0 7/8	0 3/4	0 5/8
7	1 5/8	1 7/16	1 1/4	1 1/16	0 7/8	0 3/4
8	1 7/8	1 5/8	1 7/16	1 3/16	1	0 7/8
9	2 1/8	1 7/8	1 5/8	1 3/8	1 1/16	0 15/16
10	2 3/8	2 1/16	1 13/16	1 1/2	1 1/4	1 1/16
11	2 5/8	2 5/16	2 1/16	1 11/16	1 3/8	1 1/8
12	2 7/8	2 1/2	2 3/16	1 13/16	1 1/2	1 1/4
13	3 1/16	2 11/16	2 3/8	1 15/16	1 5/8	1 3/8
14	3 5/16	2 7/8	2 1/2	2 1/8	1 3/4	1 1/2
15	3 9/16	3 1/16	2 11/16	2 1/4	1 15/16	1 5/8
16	3 3/4	3 1/4	2 13/16	2 3/8	2 1/16	1 11/16
17	4	3 1/2	3	2 9/16	2 3/16	1 3/4
18	4 3/16	3 11/16	3 3/16	2 11/16	2 5/16	1 7/8
19	4 7/16	3 7/8	3 3/8	2 7/8	2 7/16	2
20	4 11/16	4 1/16	3 1/2	3	2 1/16	2 1/8

In track work it is often necessary to ascertain the degree of a curve, though no transit is available for measuring it. The following table contains the middle ordinates of a one-degree curve for chords of various lengths:

Length of Chord in Feet.	*Middle Ordinate of a 1° Curve.*
20 ft.	1/8 in.
30 "	1/4 "
44 "	1/2 "
50 "	5/8 "
62 "	1 "
100 "	2 5/8 "
120 "	3 1/4 "

The lengths of the chords are varied so that a longer or shorter chord may be used, according as the curve is regular or not.

The table is applied as follows: Suppose the middle ordinate of a 44-feet chord is 3 inches. We find in the table that the middle ordinate of a 44-feet chord of a 1° curve is $\frac{1}{16}$ inch. Hence, the degree of the given curve is equal to the quotient of $3 \div \frac{1}{16} = 6°$ curve.

Elevation of Curves.—To counteract the centrifugal force which is developed when a car passes around a curve, the outer rail is elevated. The amount of elevation will depend upon the radius of the curve and the speed at which trains are to be run. There is, however, a limit in track elevation as there is a limit in widening gauge, beyond which it is not safe to pass.

The best authorities on this subject place the maximum elevation at $\frac{1}{4}$ the gauge, or about 8 inches for standard gauge of 4 feet $8\frac{1}{2}$ inches. The gauge on a 10° curve elevated for a speed of 40 miles an hour, should be widened to 4 feet $9\frac{1}{4}$ inches.

All curves, when possible, should have an elevated approach on the straight main track, of such length that trains may pass on and off the curve without any sudden or disagreeable lurch.

A good rule for curve approaches is the following: For each half inch or fraction thereof of curve elevation, add 30 feet or 1 rail length to the approach; that is, if a curve has an elevation of 2 inches, the approach will have as many rail lengths as $\frac{1}{2}$ is contained in 2, which is 4 times. The approach will, therefore, have a length of 4 rails of 30 feet each, or 120 feet.

The following table for elevation of curves is a compromise between the extremes recommended by different engineers. It is a striking fact that experienced trackmen never elevate track above 6 inches, and many of them place the limit at 5 inches.

Degree of Curve.	*Length of Approach.*	*Elevation.*	*Width of Gauge.*	*Speed of Trains.*
1	60 ft.	1 in.	4 ft. $8\frac{1}{2}$ in.	60 mi. per hr.
2	120 "	2 "	4 " $8\frac{1}{2}$ "	60 " "
3	150 "	$2\frac{1}{2}$ "	4 " $8\frac{3}{4}$ "	60 " "
4	180 "	$2\frac{3}{4}$ "	4 " $8\frac{3}{4}$ "	55 " "
5	180 "	3 "	4 " $8\frac{3}{4}$ "	50 " "
6	210 "	$3\frac{1}{4}$ "	4 " $8\frac{3}{4}$ "	45 " "
7	210 "	$3\frac{1}{2}$ "	4 " 9 "	40 " "
8	240 "	$3\frac{3}{4}$ "	4 " 9 "	35 " "
9	240 "	4 "	4 " 9 "	30 " "
10	270 "	$4\frac{1}{4}$ "	4 " 9 "	25 " "
11	270 "	$4\frac{1}{2}$ "	4 " $9\frac{1}{4}$ "	20 " "
12	270 "	$4\frac{3}{4}$ "	4 " $9\frac{1}{4}$ "	15 " "
13	240 "	$4\frac{1}{2}$ "	4 " $9\frac{1}{4}$ "	10 " "
14	240 "	$4\frac{1}{4}$ "	4 " $9\frac{1}{4}$ "	10 " "
15	240 "	4 "	4 " $9\frac{1}{2}$ "	10 " "
16	240 "	4 "	4 " $9\frac{1}{2}$ "	10 " "

The Elevation of Turnout Curves.—The speed of all trains in passing over turnout curves and crossovers is greatly reduced, so that an elevation of ¼ of an inch per degree is amply sufficient for all curves under 16 degrees. On curves exceeding 16 degrees, the elevation may be held at 4 inches until 20 degrees is reached, and on curves exceeding 20 degrees, ⅛ of an inch of elevation per degree may be allowed until the total elevation amounts to 5 inches, which is sufficient for the shortest curves.

The Frog.—The frog is a device by means of which the rail at the turnout curve crosses the rail of the main track. The frog shown in Fig. 2 is made of rails having the same cross-section as those used in the track. Its parts are as follows: The wedge shaped part *A* is the *tongue*, of which the extreme end *a* is the *point*. The space *b*, between the ends *c* and *d* of the rails, is the

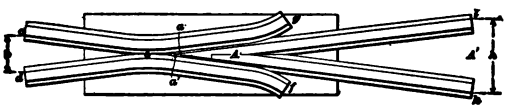


FIG. 2.

mouth, and the channel which they form at its narrowest point *e* is the *throat*. The curved ends *f* and *g* are the *wings*.

That part of the frog between *A* and *A'* is called the *heel*. The width *h* of the frog is called its *spread*. Holes are drilled in the ends of the rails, *c*, *d*. *k* and *l* to receive the bolts used in fastening the rail splices, so that the rails of which the frog is composed, form a part of the continuous track.

The Frog Number.—The number of a frog is the ratio of its length to its breadth; *i. e.*, the quotient of its length divided by its breadth.

Thus, in Fig. 2, if the length *a' l*, from point to heel of frog is 5 feet, or 60 inches, and the breadth *h* of the heel is 15 inches, the number of the frog is the quotient of 60 ÷ 15 = 4. Theoretically, the length of the frog is the distance from *a* to the middle point of a line drawn from *k* to *l*; practically, we take from *a*

to l as the distance. As it is often difficult to determine the exact point a of the frog, a more accurate method of determining the frog number is to *measure the entire length $d\,l$ of the frog from mouth to heel, and divide this length by the sum of the mouth width b and the heel width h. The quotient will be the exact number of the frog.*

For example, if in Fig. 2, the total length $d\,l$ of the frog is 7 feet 4 inches, or 88 inches, and the width h is 15 inches, and the width b of the mouth is 7 inches, then the frog number is $88 \div (15 + 7) = 4$. Frogs are known by their *numbers*. That in Fig. 2 is a No. 4 Frog.

The Frog Angle.—The frog angle is the angle formed by the gauge lines of the rails, which form its tongue. Thus, in Fig. 2, the frog angle is the angle $l\,a'\,k$. The amount of the angle may be found as follows: The tongue and heel of the frog form an isosceles triangle (see Fig. 3). By drawing a line from the point a of the frog to the middle point b of the heel $c\,d$, we form a right-angled triangle, right-angled at b. The perpendicular line $a\,b$, bisects the angle a, and, by Trigonometry, we have $\tan \frac{1}{2}\,a = \frac{b\,c}{a\,b}$. The dimensions of the frog point given in Fig. 3 are not the same as those given in Fig. 2, but their relative proportions are the same; *viz.*, the length is four times the breadth. The length $a\,b = 4$ and the width $c\,d = 1$; hence, $b\,c = \frac{1}{2}$. Substituting these values,

FIG. 3.

we have $\tan \frac{1}{2}\,a = \frac{\frac{1}{2}}{4} = \frac{1}{8} = 0.125$. Whence, $\frac{1}{2}\,a = 7°\ 7\frac{1}{2}'$ and $a = 14°\ 15'$; that is, the angle of a No. 4 frog is $14°\ 15'$.

Frog numbers run from 4 to 12, including half numbers, the *spread of the frog* increasing as the number decreases.

The Parts of a Turnout.—The several parts of a turnout are represented in Fig. 4. The distance *p f* from the P. C. of the turnout curve to the point of frog is called the *frog distance.* The radius *c o* of the turnout curve, the frog distance, the frog angle and the frog number bear certain relations to each other, which are expressed by the following formulas:

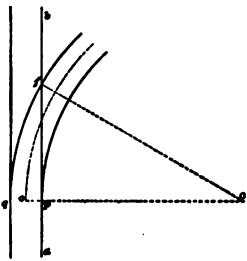


FIG. 4.

Tangent of half frog angle = gauge ÷ frog distance.

Frog number = $\sqrt{\text{radius } c\,o \div \text{twice the gauge.}}$

Frog number = 1 ÷ half the tangent of half the frog angle.

Radius *c o* = twice the gauge × square of the frog number.

Radius *c o* = (frog distance *p f* ÷ sine of frog angle) − ½ the gauge.

Radius *c o* = gauge ÷ (1 − cosine of frog angle) − ½ the gauge.

Frog distance *p f* = frog number × twice the gauge.

Frog distance *p f* = gauge *p q* ÷ tangent of half the frog angle.

Frog distance *p f* = (radius *c o* + half the gauge) × sine of frog angle.

Middle ordinate (approximate) = ¼ the gauge.

Each side ordinate (approximate) = ¾ the middle ordinate = $\frac{3}{16}$ (or .188) of the gauge.

Switch length (approximate) =

$$\sqrt{\frac{\text{throw in feet} \times 10{,}000}{\text{tan deflection for chords of 100 ft. for radius } co \text{ of turnout curve}}}$$

The tangent deflection may be obtained from the table on *pages 286-288.*

TURNOUTS FROM A STRAIGHT TRACK.

Gauge, 4 feet 8½ inches. Throw of Switch, 5 inches.

Frog Number.	Frog Angle.	Turnout Radius.	Degree of Turnout Curve.	Frog Distance.	Middle Ordinate.	Side Ordinates.	Stub Switch Length.
	° ′	Feet.	° ′	Feet.	Feet.	Feet.	Feet.
12	4 46	1,356	4 14	113.0	1.177	.883	34
11½	4 58	1,245	4 36	108.3	1.177	.883	32
11	5 12	1,139	5 02	103.6	1.177	.883	31
10½	5 28	1,038	5 31	98.9	1.177	.883	29
10	5 44	942	6 05	94.2	1.177	.883	28
9½	6 02	850	6 45	89.5	1.177	.883	27
9	6 22	763	7 31	84.7	1.177	.883	25
8½	6 44	680	8 26	80.0	1.177	.883	24
8	7 10	603	9 31	75.3	1.177	.883	22
7½	7 38	530	10 50	70.6	1.177	.883	21
7	8 10	461	12 27	65.9	1.177	.883	20
6½	8 48	398	14 26	61.2	1.177	.883	18
6	9 32	339	16 58	56.5	1.177	.883	17
5½	10 24	285	20 13	51.8	1.177	.883	15
5	11 26	235	24 32	47.1	1.177	.883	14
4½	12 40	191	30 24	42.4	1.177	.883	13
4	14 14	151	38 46	37.7	1.177	.883	11

The switch lengths in the above table merely denote the *shortest* length of *stub switch* that will at the same time form part of the turnout curve, and give 5 inches throw. *Point* or *split switches* require a throw of not more than 3½ inches, though many have a throw of 5 inches, with an equal space between the gauge lines at the heel. The heels of a split switch which occupy the same position as the toes of a stub switch, should be placed at the point where the tangent deflection or offset is 5 inches. The point where the tangent deflection is but 4½ inches will answer for many rail sections, but for those above 65 lb. per yard, 5 inches should be taken.

In the table on pages 286–288, tangent deflections for chords of *100 feet are given* for all curves up to 20°, and for a curve of

higher degree, the tangent deflection may be found by applying the formula tan deflection $= \frac{c^2}{2R}$.

In complicated track work where space is limited, curves must be chosen to meet the existing conditions, and not with reference to particular frog angles, in which case the frogs are called special frogs, and are made to fit the particular curve used. The determination of the frog distance, switch length and frog angle may be understood by referring to Fig. 5.

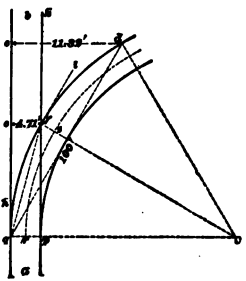


FIG. 5.

Let the main track $a\ b$ be a straight line; the gauge $p\ q$ = 4 feet 8½ inches (= 4.71 feet); the degree of the turnout curve = 13°; the chord $q\ d$ = 100 feet; $c\ d$ = the tangent deflection of the chord $q\ d$, and $p\ f$ = the frog distance. From the table on page 288 we find the tangent deflection for a chord 100 feet long of a 13° curve is 11.32 feet.

Then, from Fig. 5, we have the proportion

$$c\ d : e\ f :: \overline{q\ c}^2 : \overline{q\ e}^2.$$

Now, in curves of large radius $q\ c$ and $q\ d$ are assumed to be equal. Also, $q\ e = p\ f$, the frog distance, and substituting these equivalents we have the proportion

$$c\ d : e\ f :: \overline{q\ d}^2 : \overline{p\ f}^2.$$

Substituting the above given quantities in the proportion, we have

$$11.32 : 4.71 :: 100^2 : \overline{p\ f}^2;$$

whence, $\overline{p\ f}^2 = \frac{100^2 \times 4.71}{11.32}$, and

frog distance, $p\ f$ = 64.5 feet.

If the space between the gauge lines at the heels of a split switch be taken at 5 inches = 0.42 of a foot, the distance from the P. C. of the turnout curve to the heel of the switch may be found as follows:

In Fig. 5, let h, the tangent off-set at the heel of the switch = 0.42 of a foot, we have the proportion

$$c\,d : h :: \overline{q\,d}^2 : \overline{q\,h}^2,$$

and substituting known values, we have

$$11.32 : 0.42 :: 100^2 : \overline{q\,h}^2,$$

whence, $$\overline{q\,h}^2 = \frac{10{,}000 \times 0.42}{11.32} = 371.02, \text{ and}$$

$$q\,h = 19.26 \text{ feet.}$$

This locates the heel of a *split switch* and the toe of a *stub switch*.

The *frog angle* is the angle $k\,f\,l$ (see Fig. 5) formed by the gauge line of the main rail $f\,k$ and the tangent to the outer rail $q\,f$ of the turnout curve at the point where the two rails intersect. This angle is equal to the central angle $q\,o\,f$. The arcs $q\,f$ and $r\,s$ are assumed to be of the same length. The turnout curve being 13°, the central angle for a chord of 1 foot is $\frac{13 \times 60}{100} =$ 7.8′, and the central angle for 64.5 feet, the *frog distance*, is 7.8′ × 64.5 = 8° 23′, the frog angle for a 13° curve. By this process the frog distance, switch length and frog angle may be calculated for curves of any radius.

To Lay Out a Turnout from a Curved Main Track.—There are two cases:

Case I.—When the two curves deflect in opposite directions, illustrated in Fig. 6, and

Case II.—When the two curves deflect in the same direction, *illustrated* in Fig. 7.

In Fig. 6, the curve $a\,b$ is 3° 30′, and it is proposed to use a ' frog. By reference to the table on page 304 we find that

the degree of curve corresponding to a No. 8 frog is 9° 31′. Accordingly, we use a turnout curve *a e*, whose degree when added to the degree of curve of the main track, shall equal the degree required for a No. 8 frog; *i. e.*, we use a 6° turnout curve, which is within one minute of the required degree, and close enough for practical purposes. We know that for curves of moderate radii, *i. e.*, from 1° up to 12°, the tangent deflections or off-sets increase as the degree of the curve. That, is, the tangent deflection of a 2°, 4° and 6° is two, four and six times, respectively, that of a 1° curve. In the accompanying cuts illustrating the location of frogs and switches, each curve is represented by two lines indicating the rails, whereas only the center lines of the curves are run in on the ground. In Fig. 6, the line *c d* is tangent to the center lines of the curves. These center lines do not appear in the cut.

Now, if in Fig. 6, a tangent *c d* be drawn at *c*, the point common to the center lines of the curves, the sum of the deflections of both curves from the common tangent will be equal to the tangent deflection of a 9° 30′ curve from a straight line.

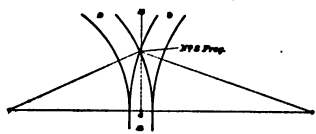


FIG. 6.

Accordingly, to find the frog distance for a 6° turnout curve from a 3° 30′ curve, the curves being in opposite directions, as shown in Fig. 6, we find the tangent deflection of a 9° 30′ curve for a chord of 100 feet. This deflection is 8.28 feet, as given in the table on page 288.

Assuming the gauge of track to be standard, *viz.*, 4 ft. 8½ in. = 4.71 ft. and denoting the required frog distance by x, we have the following proportion:

$$8.28 : 4.71 :: 100^2 : x^2,$$

whence, $$x^2 = \frac{10{,}000 \times 4.71}{8.28} = 5{,}688.4, \text{ and}$$

frog distance, $$x = 75.42 \text{ ft.}$$

We use the tangent deflection for a 9° 30′ curve, which is practically the same as for a 9° 31′ curve, and so save the labor of a *calculation*, which will not appreciably affect the result.

We locate the heel of the switch in the same way, using for the second term of the proportion, 0.42 foot, the distance between the gauge lines at the heel, instead of 4.71 feet, the gauge of the track.

In Fig. 7, which comes under Case II, both curves deflect in the same direction, and the rate of their deflection from each other is equal to the rate of the deflection of a curve whose degree is equal to the difference of the degrees of the two curves, from a tangent.

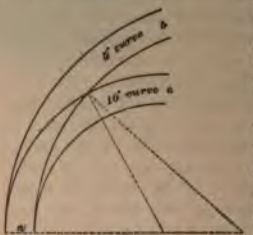


FIG. 7.

Let the main track curve $a\,b$ be 5°, and the turnout curve $a\,c$ be 10°. Then, the rate of deflection or divergence of the 10° curve from the 5° curve is equal to the divergence of a (10°—5°) 5° curve from a straight track or tangent.

Accordingly, we find in the table on page 287, the tangent deflection for a 5° curve for a chord of 100 feet = 4.36 feet. Denoting the required *frog distance* by x, we have the following proportion:

$$4.36 : 4.71 :: 100^2 : x^2,$$

whence, $$x^2 = \frac{10{,}000 \times 4.71}{4.36} = 10{,}802{,}8, \text{ and}$$

frog distance $$x = 103.9 \text{ feet.}$$

Distances are not calculated nearer than to tenths of a foot.

How to Lay Out a Switch.—In laying out a switch, locate the frog so as to cut the least possible number of rails. Where there is some latitude in the choice of location, the P. C. of the turnout curve can be located, so as to bring the frog near the end of a rail.

To do this, take from the table on page 304 the frog distance corresponding to the number of the frog to be used. Locate approximately the P. C. of the turnout curve, and measure from it *the main* track rail, the tabular frog distance. If this *he frog* point near the end of the rail, the P. C. of the

turnout curve may be moved so as to require the cutting of *but one* main track rail. Measure the total length of the frog and deduct it from the length of the rail to be cut, marking with red chalk on the flange of the rail the point at which the rail is to be cut. Measure the width of the frog at the heel and calculate the distance from the heel to the theoretical point of frog. For example, if the width of the frog at the heel is 8½ inches, and a No. 8 frog is to be used, the theoretical distance from the heel to the point of frog is $8.5 \times 8 = 68$ inches $= 5$ feet 8 inches. Measure off this distance from the point marking the heel of the frog. This will locate the point of frog, which should be distinctly marked with red chalk on the flange of the rail. It is a common practice to make a distinct mark on the web of the main track rail, *directly opposite* the point of frog. This point being under the head of the rail it is protected from wear and the weather. The P. C. of the turnout curve is then located by measuring the frog distance from the point of frog. From the table on page 304 we find the frog distance for a No. 8 frog is 75.3 feet, and the switch length, *i. e.*, the distance from the P. C of the turnout curve to the heel of the split switch or toe of the stub switch, is 22 feet.

If a *stub switch* is to be laid, make a chalk mark on both main track rails on a line marking the center of the head block. A more permanent mark is made with a center punch. Stretch a cord touching these marks, and drive a stake on each side of the track, with a tack in each. This line should be at right angles to the center line of the track, and the stakes should be far enough from the track not to be disturbed when putting in switch ties. Next, cut the switch ties of proper length; draw the spikes from the track ties, three or four at a time, and remove them from the track, replacing them with switch ties, and tamping them securely in place. When all the long ties are bedded, cut the main track rail for the frog, being careful that the amount cut off is just equal to the length of the frog. If, by increasing or decreasing the length of the lead 5 per cent., cutting a rail can be avoided, do not hesitate to do so, especially for frogs above No. 8.

Use *full length* rails (30 feet) for moving or switch rails, and *be careful to leave* a joint of proper width at the head chair.

Spike the head chairs to the head block so that the main track rails will be in perfect line. Spike from 8 to 10 feet of the switch rails to the ties, and slide the cross rods on to the rail flanges, spacing them at equal intervals. The cross rods are placed between the switch ties which should not be more than 15 inches from center to center of tie. The switch ties, especially those under the moving rails, should be of *sawed oak timber*. Southern pine is a good second choice. Attach the connection rod to the head rod and to the switch stand. With these connections made, it is an easy matter to place the switch stand so as to give the proper throw of the switch.

It is common practice to fasten the switch stand to the head block with track spikes, but a better fastening is made with bolts. The stand is first properly placed and the holes marked and bored, and the bolts passed through from the under side of the head block. This obviates all danger of movement of the switch stand in fastening, which is liable to occur when spikes are used, and insures a *perfect throw*.

The use of track spikes is quite admissible when holes are bored to receive them, in which case a half-inch augur should be used for standard track spikes. The switch stand should, when possible, be placed facing the switch, so as to be seen from the engineer's side of the engine—the right-hand side.

Next stretch a cord from *a*, Fig. 8, a point on the outer main track rail opposite the P. C. of the turnout curve, to *b*, the point of the frog. This cord will take the position of the chord of the arc of the outer rail of the turnout curve. Mark the middle point *c* and the quarter points *d* and *e*. Whatever the degree of the turnout curve, the distance from the middle point *c* of the chord to the arc *a b* is 1.18 feet, and the distances from the quarter points *d* and *e* are 0.88 foot; hence, at *c* lay off the ordinate 1.18 feet, and at both *d* and *e* the ordinate 0.88 foot, three quarters of the middle ordinate. These off-sets will mark the gauge line of the rail *a b*. Add to these off-sets the distance from the gauge line to outside of the rail flange, and mark the points on the switch

Fig. 8.

ties. Spike a lead rail to these marks and place the other at easy track gauge from it. Spike the rails of the turnout as far as the point of frog to exact gauge, unless the gauge has been widened owing to the sharpness of the curve. Beyond the point of frog the curve may be allowed to vary a little in gauge to prevent a kink showing opposite the frog. In case the gauge is widened at the frog, increase the guard rail distance an equal amount. For a gauge of 4 feet 8½ inches, place the side of the guard rail which comes in contact with the car wheels at 4 feet 6⅝ inches from the gauge line of the frog. This gives a space of 1⅜ inches between the main rail and the guard rail.

In case the gauge is widened ¼ or ½ inch, increase the guard rail distance an equal amount.

When the turnout curve is very sharp, it will be necessary to curve the switch rails, to avoid an angle at the head block. The lead rails should be carefully curved before being laid, and great pains taken to secure a perfect line.

If a *point*, or *split*, *switch* is to be laid, the order of work is nearly the same. The same precautions must be taken to avoid the unnecessary cutting of rails, with the additional precaution of keeping the switch points clear of rail joints, as the bolts and angle splices will prevent the switch points from lying close to the stock rails. As already stated, these conditions can usually be met where there is some range in the choice of the location of the switch. Where there is none, the main track rails must be cut to fit the switch.

Having located the point of frog, the P. C. of the turnout curve and the heel line of the switch, measure back from the heel line a distance equal to the length of the switch rails, and place on the flange of each rail a chalk mark to locate the ends of the switch points. This will also locate the head block. Prepare switch ties of the requisite number and length, and place them in the track in proper order. As in the case of stub switches, see to it that all long switch ties are in place before cutting the rail for placing the frog; also, that the ends of the lead rails, with which the switch points connect, are exactly *even, otherwise*, the switch rods will be skewed, and the switch *will not work* or fit well. Fasten the switch rods in place.

being careful to place them in their proper order, the head rod being No. 1. Each rod is marked with a center punch, the number of the punch marks corresponding to the number of the rod.

Couple the switch points with the lead rails and place the sliding plates in position, securely spiking them to the ties. Connect the head rod with the switch stand, and close the switch, giving a clear main track.

Adjust the stand for this position of the switch and bolt it fast to the head block. Next, crowd the stock rail against the switch point so as to insure a close fit, and secure it in place with a rail brace at each tie; then continue the laying of the rails of the turnout.

If there is no engineer to lay out the center line of the turnout, the section foreman can put in the lead from ordinates, as explained in Fig. 8. In modern railroad practice, however, most track work is done under the direction of an engineer, in which case the center line of the turnout is located with a transit. This insures a correct line and expedites work. For ordinary curves, center stakes at intervals of 50 feet are sufficient, excepting between the P. C. of the turnout and the point of frog, where there should be a center stake at each interval of of 25 feet. Place a guard rail opposite the point of frog on both main track and turnout. The guard rail should be 10 feet in length; this is an economical length for cutting rails, as each full length rail makes three guard rails.

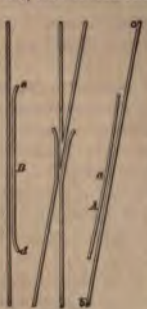


FIG. 9.

Two styles of guard rails are shown in Fig. 9. That shown at *B* is in general use, but the style shown at *A* is growing in favor. The latter is curved throughout its entire length. At its middle point *a*, directly opposite the point of frog, the guard rail is spaced 1⅞ inches from the gauge line of the turnout rail *b c*. From this point the guard rail diverges in both directions, giving at each end a flange way of 4 inches. This allows the wheels full play, excepting at the point of frog, where the guard

rail is exactly adjusted to the track gauge, and holds the wheels in true line, preventing them from *climbing* or *mounting* the frog. The style of guard rail shown at *B*, though still much used, has two objectionable features, *viz.*, first, the abruptly curved ends *d* and *e* often receive an almost direct blow from the wheel flanges, which causes a car to lurch violently; and second, the flange-way of uniform width, though proper for the main track when straight, as in Fig. 9, is unsuited for sharp curves on either a main track or a turnout, as it compels the wheels to follow a curved line; whereas, the normal position of the wheel base of each truck is that of a chord of or a tangent to the curve. These two defects alone, produce what is known as a *rough riding* frog, even though the frog is well lined and ballasted.

Location of Crotch Frog.—A *crotch* or *middle* frog is a frog placed at the point where the outer rails of both turnouts of a three-throw switch cross each other. When both turnouts are of the same degree, the crotch frog comes midway between the main track rails. Its location and angle may be determined as follows: Let the turnout curves *A* and *B*, Fig. 10, be each 9° 30′, uniting with the main track *C* by a three-throw switch. Let *a* be the P. C. common to both curves, and *b*, the location of the crotch or middle frog.

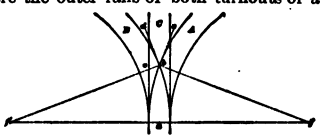


FIG. 10.

It is evident that the point of the crotch frog should be exactly midway between the gauge lines of the main track rails, and if the gauge is 4 feet 8½ inches = 4.71 feet, the point of the crotch frog will be $\frac{4.71}{2}$ = 2.35 feet from each rail. Now, the problem is to find the frog distance from *a*, the P. C., to the point *c*, where the tangent deflection will equal 2.35, or half the gauge. From the table on page 288, we find the tangent deflection of a 9° 30′ curve is 8.28 feet. Applying the principle explained in connection with Fig. 5 and letting *x* represent the required *frog distance*, we have the following proportion:

$$8.28 : 2.35 :: 100^2 : x^2;$$

whence, $$x^2 = \frac{100^2 \times 2.35}{8.28} = 2.838.2 \text{ feet},$$

and $$x = 53.3 \text{ feet, nearly,}$$

the required frog distance.

Now, there are two curves starting at the common point a; the outer rails intersect at b, and the angle $d\ b\ e$, formed by tangents drawn to the point of intersection is the angle of the crotch, or middle frog. The angle is equal to the sum of the angles $a\ f\ b$ and $a\ f'\ b$; that is, equal to double the central angle of either curve between the P. C. and the point of intersection b. The degree of the curve is $9° 30' = 570'$ and the central angle or total deflection for each foot is $\frac{570'}{100} = 5.7'$, and for the frog distance of 53.3 feet, the central angle is $53.3 \times 5.7 = 303.8' = 5° 03.8'$. The angle of the crotch frog is double this angle; *i. e.*, $5°$

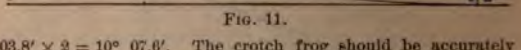


FIG. 11.

$03.8' \times 2 = 10° 07.6'$. The crotch frog should be accurately located and spiked in place before the lead rails are placed.

The one objection to the three-throw switch is the open joint at the head block, the inevitable attendant of the stub switch, but its advantages are so great that it will continue to be used, especially in yard service.

Cross-over Tracks.—A *cross-over* is a track by means of which a train passes from one track to another. The tracks united are usually parallel, as are the tracks of a double track road. Such a cross-over is shown in Fig. 11. The tracks $a\ b$ and $c\ d$ are 13 feet apart from center to center, which is the standard distance for double tracks. The cross-over consists of two turnout curves, $e\ f$ and $g\ h$. These curves are usually, though not necessarily, of the same degree. The curves terminate at the points of frog f and h, between which the track $f\ h$ is a tangent. The *essential point in* laying out a cross-over is to so place the frogs *that the connecting* track shall be tangent to both curves. In *suppose the frogs* are No. 9, requiring 7° 31′ turnout curves.

From the table on page 304, we find the required frog distance is 84.7 feet. and the switch length 25 feet. As previously noted, if there is considerable range in choice of location, the frogs can be so placed as to largely avoid the cutting of rails; but usually cross-overs are required at certain precise places, and the rails must be cut as occasion demands. Having located the point of frog at *f* we determine the point of the next frog at *h*, as follows: A No. 9 frog is one which spreads 1 inch in width to every 9 inches in length, and as the track between the frog points is straight, the distance *f h* between these points will be as many times 9 inches as is the space *k* between the tracks at the frog point *f*. The main track centers are 13 feet apart, making the space between the gauge lines of the inside rails 8 feet 3½ inches. As it is the rail *l* of the turnout which joins the second frog at *h*, we subtract the gauge, 4 feet 8½ inches, from 8 feet 3½ inches, leaving 3 feet 7 inches, the distance *k*, between the gauge line of the rail *l*, opposite the frog point *f*, and the gauge line of the nearest rail of the track *c d*. This distance multiplied by 9 inches will give the distance from the frog point *f* to the frog point *h*; 3 feet 7 inches = 43 inches, 43 × 9 = 387 inches = 32 feet 3 inches. Accordingly having located the point or frog *f*, we mark a corresponding point on the nearest rail of the opposite track. From this point we measure along the rail the distance 32 feet 3 inches, locating the second frog point *h*, and again the frog distance 84.7 feet to the P. C. of the second turnout curve at *g*.

If frogs of different numbers, say 7 and 9, were to be used, the distance between the frogs is found as follows:

As the No. 7 frog spreads 1 inch in 7 inches, and the No. 9 frog 1 inch in 9 inches, the two will together spread 2 inches in 7 + 9 = 16 inches, or 1 inch in 8 inches. Now, if the rails to be united are 3 feet 7 inches, or 43 inches, apart, as in the previous problem, the distance between the frog points will be 43 × 8 = 344 inches = 28 feet 8 inches.

In locating cross-over tracks, regard should be paid to the direction in which the bulk of the traffic moves, and the cross-over tracks should be so placed that loaded cars will be backed, not pushed, from one track to the other.

At all stations on double track roads there should be a cross-*over to facilitate* the exchange of cars and the making up of trains

PERPETUAL CALENDAR—1797-1904.

3	4	5	6	0	1	2
June.	Sept. Dec.	April. July.	Jan. Oct.	May.	Aug.	Feb. Mar. Nov.
1797	1798	1799	1800	1801	1802	1803
	1804	1805	1806	1807		1808
1809	1810	1811		1812	1813	1814
1815		1816	1817	1818	1819	
1820	1821	1822	1823		1824	1825
1826	1827		1828	1829	1830	1831
	1832	1833	1834	1835		1836
1837	1838	1839		1840	1841	1842
1843		1844	1845	1846	1847	
1848	1849	1850	1851		1852	1853
1854	1855		1856	1857	1858	1859
	1860	1861	1862	1863		1864
1865	1866	1867		1868	1869	1870
1871		1872	1873	1874	1875	
1876	1877	1878	1879		1880	1881
1882	1883		1884	1885	1886	1887
	1888	1889	1890	1891		1892
1893	1894	1895		1896	1897	1898
1899	1900	1901	1902	1903		1904
Sun.	Mon.	Tues.	Wed.	Thur.	Fri.	Sat.
1	2	3	4	5	6	7
8	9	10	11	12	13	14
15	16	17	18	19	20	21
22	23	24	25	26	27	28
29	30	31	32	33	34	35
36	37	38	39	40	41	42
43	44					

By means of the table given on opposite page, the day of the week corresponding to any date between 1797 and 1904, inclusive, may be readily found. Before every leap year, there is a blank space. To find the day of the week on which January 1 of any year fell, find that year in the table; glance down the column containing that year, and the day of the week at the foot of the column will be the day of the week required. Thus, to find on what day of the week January 1, 1895, fell, we find under 1895 in the table, Tuesday. For leap years, we look for day of week under the blank space before the year. Thus, January 1, 1896, fell on Wednesday, Wednesday being in the column containing the blank space before 1896. To find the day of the week for any other date, add (mentally) to the day of the month the first number under the day of the week which is contained in the column containing the year of the century; to this sum, add the number above the month at the top of the table. Find the number thus obtained in the columns of figures under the days of the week; the day of the week at the head of the column containing this number will be the day required. Thus, to find on what day of the week September 10, 1813, fell, we find 1813 in the table. The number under the day of the week in the column containing 1813 is 6, and the number above September at the top of the table is 4. Hence, $10 + 6 + 4 = 20$. The day of the week above 20, in the lower part of the table, is Friday.

For dates in January and February of leap years, take one day less, or add the number beneath the day of the week under the blank space preceding the year. Thus, for February 12, 1896, we have $12 + 4 + 2 = 18$, and the day of the week above 18 is Wednesday.

The table may also be used for fixing dates. Thus, Thanksgiving Day is the last Thursday in November; on what day of the month will it fall in 1897? Since the earliest day on which it can fall is the 24th, we find on what day of the week November 24 falls, and then count ahead to Thursday. Referring to the table, $24 + 6 + 2 = 32$; the day of the week above 32 is *Wednesday*, and since Thursday is one day later, it follows that *Thanksgiving* Day in 1897 falls on the 25th.

of the week July 25, 1872, fell, and then count backwa
Monday. Thus, $25 + 3 + 5 = 33$; the day of the week ab
is Thursday. Since Thursday is three days after Monda
$25 - 3 = 22$, the event occurred on July 22.

Technical Subjects Taught by Mail.

Our correspondence method of instruction offers to workingmen, young men and others who cannot leave home to attend college, the best opportunity to educate themselves. The student loses no time from work but attends school in his own home when it is most convenient for him to study. Each student is a class by himself; he is directed and assisted in his studies by the ablest Instructors. The instruction is thorough and practical. Our methods are successful. Our Schools have received the endorsement of leading educators, engineers and technical journals. We can refer to students in every part of the United States, Canada and Mexico, and in many other countries.

We furnish instruction by mail in

32 COURSES.	PRICES In Installments.	In Advance.
1. Complete Mechanical	$ 45.00	$ 40.00
2. Mechanical Drawing	30.00	25.00
3. Stationary Engineers'	35.00	30.00
4. Marine Engineers'	35.00	30.00
5. Locomotive Engineers'	35.00	30.00
6. Mechanical-Electrical	65.00	60.00
7. Electrical Power and Lighting	35.00	30.00
8. Complete Architectural	65.00	60.00
9. Architectural Drawing and Designing	45.00	40.00
10. Architectural Drawing	30.00	25.00
11. Sheet Metal Pattern Drafting	55.00	50.00
12. Tinsmiths' Pattern Cutting	40.00	35.00
13. Sanitary Plumbing, Heating and Ventilation	45.00	40.00
14. Sanitary Plumbing and Gas Fitting	35.00	30.00
15. Heating and Ventilation	35.00	30.00
16. Civil Engineering	185.00	175.00
17. Railroad Engineering	50.00	45.00
18. Surveying and Mapping	35.00	30.00
19. Bridge Engineering	50.00	45.00
20. Municipal Engineering	60.00	55.00
21. Hydraulic Engineering	60.00	55.00
22. Mathematics and Physics	22.00	20.00
23. Full Mining	65.00	60.00
24. Complete Coal Mining	45.00	40.00
25. Metal Mining	45.00	40.00
26. Mine Mechanial	45.00	40.00
27. Short Coal Mining	32.00	27.00
28. Metal Prospectors'	Not sold on Installments	22.00
29. Complete Commercial	45.00	40.00
30. Complete Stenographic	30.00	25.00
31. *Bookkeeping* and Business Forms	30.00	25.00
32. *English* Branches	30.00	25.00

Method of Teaching.

Courses in The International Correspondence Schools may be commenced at any time. All that the student is required to know in order to enroll is how to read and write English. When paid in full a Scholarship is non-forfeitable; in other words, a student may continue his studies with us until he has completed all the subjects that are embraced in his scholarship, and has been awarded his Diploma. The scholarship is transferable, and the student need not leave home in order to study—he may reside anywhere in the world so long as it is possible for him to be reached through the mails. No books need be purchased, as the Instruction and Question Papers are all the books that he needs. These are written in clear and concise language, as free as possible from technicalities, and begin at the beginning of each subject. They explain principles and details and how to apply them. All unnecessary matter is omitted. Whenever the student desires special information it is supplied him freely, and in such detail that he cannot fail to understand it; blanks upon which to make requests for such information are furnished by the Schools. The tuition fee covers every expense except the cost of such writing materials and drawing instruments as the student may require, and the postage on his matter to us; we pay all postage on matter which we send to the students.

Testimonials.

On application we will send to any address a Book of Testimonials containing the hearty and unqualified endorsements of the Schools and the system of instruction, which we have received from our students.

Further, to persons thinking of enrolling as students, we will send the name and address of a student living in the same neighborhood whom they may consult or correspond with, as to the practicability of the method of teaching and the value of the instruction imparted.

Commercial Standing.

The Colliery Engineer Co., proprietors of the International Correspondence Schools, is incorporated under the laws of Pennsylvania, and has a Credit Rating in the Mercantile Agencies of *R. G. Dun & Co.* and Bradstreet's, of from $300,000 to $500,000.

Bound Volumes of Instruction Papers.

The prices of the Scholarships include a complete set of bound volumes of the Instruction Papers, Question Papers and Drawing Plates used in the course, with the Keys and Tables and an abstract of the formulas, conveniently arranged for reference and fully indexed, bound in half leather.

Payment by Installments.

When paid on the installment plan, the first payment is either $5, or $10, according to the course selected, and the balance is payable in monthly installments of $5.00. If students cannot pay the installments as they become due, upon written request they will be granted such extension of time as may be necessary.

Instruction is also given in the following

SEPARATE STUDIES.

Arithmetic	$5.00	Double Entry Bookkeeping	$15.00
Spelling	5.00	How to Change from S. E. Bookkeeping to D. E. Bookkeeping and *vice versa*	5.00
Penmanship	5.00	Stenography	15.00
Grammar	5.00	Algebra	5.00
Letter Writing	5.00	Geometry and Trigonometry	5.00
Geography	5.00		
U. S. History	5.00		
U. S. Civil Government	5.00		
Single Entry Bookkeeping	5.00		

Separate studies must be paid for in advance.

For Whom Designed.

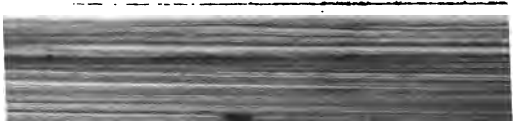
Our correspondence method of instruction is designed to meet the requirements of all who, for any cause, cannot attend schools or colleges, but desire to gain an education that they may fit themselves for advancement. Young men entering the trades or engineering professions, workingmen in all industries, foremen and superintendents of plants, officials of companies, office men, and all who desire to qualify themselves for responsible and remunerative positions can educate themselves in these Schools, *without loss of* time from work.

Free Circulars of Information.

Those interested, either for themselves or their friends who intend enrolling, should send to us for one or more of the following circulars of information, which will be sent free of charge to any address on application.

	Pages.
The Correspondence School of Mines	112
The Correspondence School of Mechanics	56
The Correspondence School of Steam Engineering	92
The Correspondence School of Electricity	112
The Correspondence School of Architecture	56
The Correspondence School of Civil Engineering	104
The Correspondence School of Plumbing, Heating and Ventilation	92
The Correspondence School of Sheet Metal Pattern Drafting	50
The Correspondence School of Bookkeeping and Stenography	56
The Correspondence School of English Branches	72

Address

The International Correspondence Schools,

SCRANTON, PA.